# COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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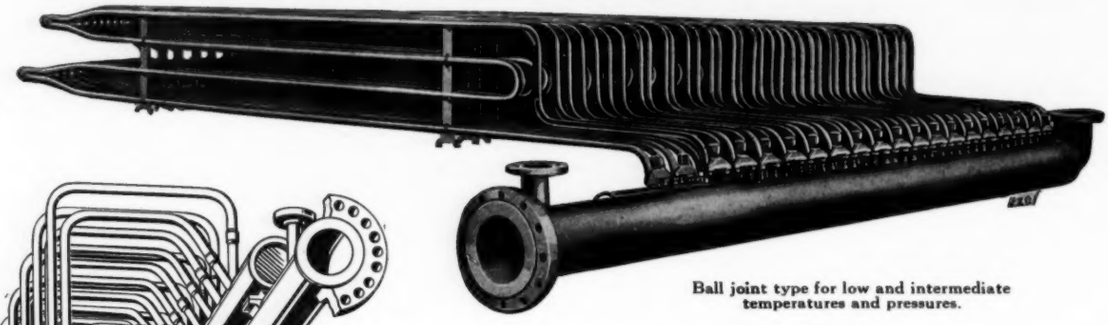

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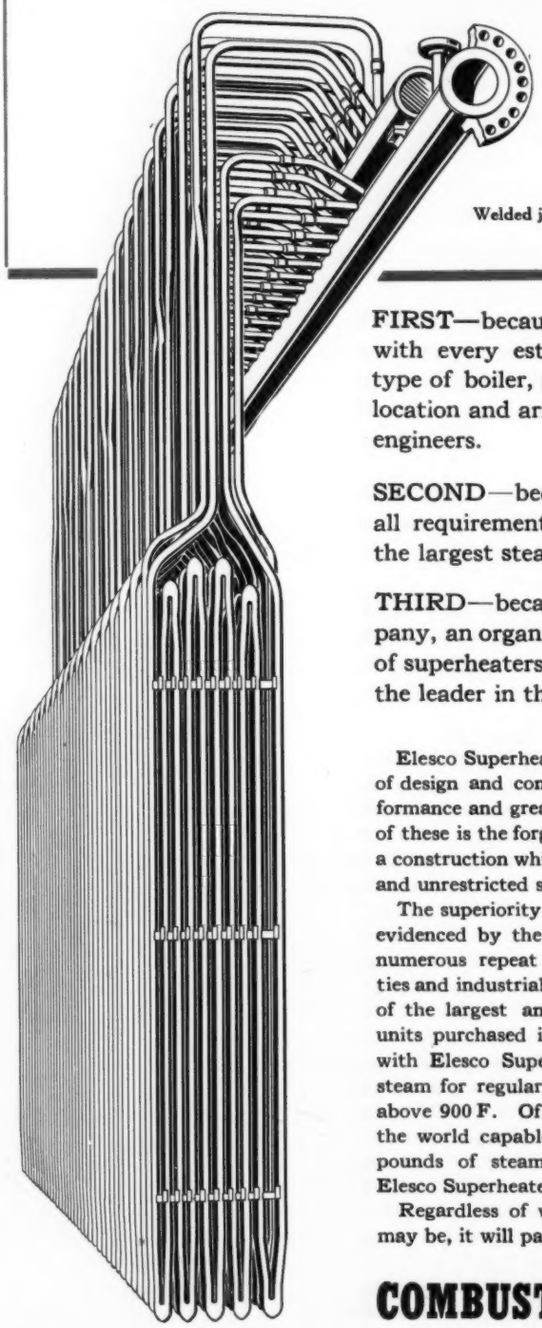

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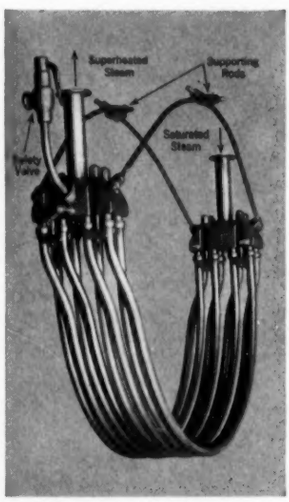


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# COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME NINE NUMBER ELEVEN

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# EDITORIAL

## Sniping at the Professional Engineers

Professional engineers have made an excellent start in obtaining public recognition through registration and license laws in forty states and further progress is being achieved toward extending their influence in the engineering affairs of various communities. However, in order that the ground thus gained may not be lost through legislation inimical to their interests, constant vigilance is necessary.

For instance, during the last session of the New York State Legislature several bills, sponsored chiefly by labor groups, were introduced to break down this professional status. One of these aimed at classifying engineers and architects as laborers and thus bringing them within the jurisdiction of existing labor boards; a second proposed to license all stationary engineers of ten years' experience as professional engineers; a third would have restricted the scope of professional engineering and nullified the present protection of the term "engineer"; and a fourth was directed at restricting the engagement of consulting engineers on public work. It is likely that similar proposals were introduced before the legislative bodies of other states.

Legislators, as a rule, are responsive to the pressure of organized groups, which can be counteracted only by vigorous expressions of opinion by those adversely affected. It therefore behooves engineers to take a closer interest in such proposals by contacting their representatives individually or through their engineering societies. Credit for killing the bills mentioned belongs chiefly to the New York State Society of Professional Engineers.

## Smoke Abatement Truisms

The U. S. Bureau of Mines has just issued a circular containing selections from the writings of the late O. P. Hood on "Smoke Abatement." Those who knew Dr. Hood during the many years he was chief mechanical engineer of the Bureau will recall his simple philosophy and ability to analyze problems in a clear and comprehensive way. This was particularly true of his views on smoke abatement, a subject to which he had given much attention. In answer to why it is that smoke abatement efforts do not bring more satisfactory results, in 1926 he wrote:

"The main reason is an incorrect estimate of the kind and size of the job. When an industrial community realizes that keeping the air clean is a project of similar magnitude to keeping streets clean, providing clean water, removing city waste or guarding the moral atmosphere, then there may be real hope of success. We usually think of the project in too small terms. It is thought of as a job or two, a small appropriation, to be abolished in a fit of economy and reestablished under pressure of a vigorous minority. Continuous, never-ceasing effort is required to clean up a smoky city. A few months of relaxation and the situation slips back some years of effort."

How well this reflects the experiences of many communities. Again, he wrote:

"The past twenty years have shown that the best method of dealing with the problem is to have an engineering service that can lead by demonstration and good advice, telling people just what to do and seeing that it is done. This means leadership, rather than spotting and finding offenders. It means control of all new installations and a program of continued education."

These last observations of Dr. Hood were made eight years ago. Through the installation of effective dust recovery equipment many of the large plants are now successfully coping with this problem and the number of stokers installed during the past few years has materially helped the small plant situation and also, to some extent, the domestic problem. Progress is being made in many communities but the advice of Dr. Hood still holds.

## Power Plant Theses

For several months past numerous letters, some addressed to this magazine and others to its parent company, have been received from engineering seniors seeking information and assistance in the preparation of their theses dealing with power plant design. Undoubtedly other papers and equipment manufacturers have received similar requests. It is logical that students should look to such sources for practical information and it is always a pleasure to render them assistance.

From a perusal of these letters, however, one is led to the conclusion that in many cases the students lack a clear conception of the basic conditions they have selected, particularly with reference to current practice, and in other cases the problems assigned or chosen are too ambitious to be solved by the limited knowledge of the average senior. Many of the letters indicate a failure to grasp the fundamental approach to a problem in power plant design and the sequence of the steps involved. This is especially noticeable in letters from students in some of the smaller schools.

It perhaps may be expecting too much to assume that students should possess such knowledge inasmuch as their studies can serve only as the basis of an engineering career; but the purpose of a thesis is to evaluate the student's ability to initiate and successfully handle a complete problem based on application of the knowledge he has gained in his university work, supplemented by such information as he is able to gather from the field.

On this basis, and in view of many of the letters received, it would seem that instructors might be more discriminating in assigning theses on power plant design. Where such assignments are made the problem should be kept as simple as possible, so that the student may have the experience of executing a complete preliminary design in all its phases.

# Utilizing Waste Fuel at ANTIOCH POWER CO.

By C. M. GARLAND,
Consulting Engineer, Chicago

**This 10,000-kw station, supplying electricity to the town of Linton, Ind., and to several nearby coal mines, burns slurry from coal-washing plants on traveling grate stokers. This fuel contains a large amount of clay, from 18 to 24 per cent moisture, 20 to 30 per cent ash, and has a heating value of 8000 to 9000 Btu per lb; it ranges in size from $^1/_{16}$ in. down. Despite this, the boilers have been operated up to 300 per cent rating. The mine load, because of the stripping shovels, imposes very severe peaks. Although second-hand equipment was employed, the maintenance during three years operation has been practically negligible. The water rate approximates 23 lb per kw-hr sent out and the evaporation averages 5.82 lb per pound of fuel.**

DURING the past twenty years the writer has at one time or another utilized a number of low grade fuels and waste products as fuel for the generation of steam and power. Included in these were gob from picking tables running as high as 4.5 per cent sulphur and 35 per cent ash, lignites running from 6000 to 8000 Btu per lb, "bug dust" from coal mines with different calorific values, wood refuse, bark from hemlock logs, and, in the case of the plant of the Antioch Power Company, the slurry from coal washers. It is, indeed, the slurry from the washing plant that made possible the power generating station of the Antioch Power Company.

This plant is owned by the Sherwood-Templeton interests, and was built to furnish power to a number of their mines located around Linton, Indiana, to the City of Linton, and to any other municipalities within the territory that might care to avail themselves of this source of supply. The generating station was located on the property of the Friar Tuck mine, which is a strip mine and on a private railway siding, adjacent to a large field of slurry which had been accumulating for a number of years. The station is about a quarter of a mile from the coal washing plant of the Friar Tuck mine, and a few hundred feet farther from the New Hope mine, a shaft mine located across the railroad tracks from the Friar Tuck. The fuel supply is a mixture of the slurries from the two mines, sent to the generating station through flumes.

R. H. Sherwood, President of the Sherwood-Templeton Coal Company, had believed for a number of years that the slurry from washing plants might some day become of value, and when it was pointed out to him that we were using "bug dust" from another mine for the generation of power he requested an opinion on the utilization of the slurry. We informed him that the probabilities were that the slurry could be utilized, and recommended making some preliminary tests on a traveling grate type of stoker as this seemed to be the most feasible way of utilizing material containing such a high percentage of moisture. The preliminary tests indicated that the fuel could be burned, and the writer's company was commissioned to design the installation. Construction was started in the winter of 1934 and the plant was placed in operation in October 1935.

Due to Mr. Sherwood's foresight and belief that the slurry from washing plants would ultimately become of value, he had adopted the practice of storing this slurry from his different washing plants in fields which were surrounded by bales of straw. The slurry from the plant was run into these enclosures, the water ran out and the slurry was deposited. In the case of the Antioch plant, a slurry field of about 15 acres had been formed by packing straw against a chicken-wire fence in the bottom land adjacent to a hill, the straw forming one side of the

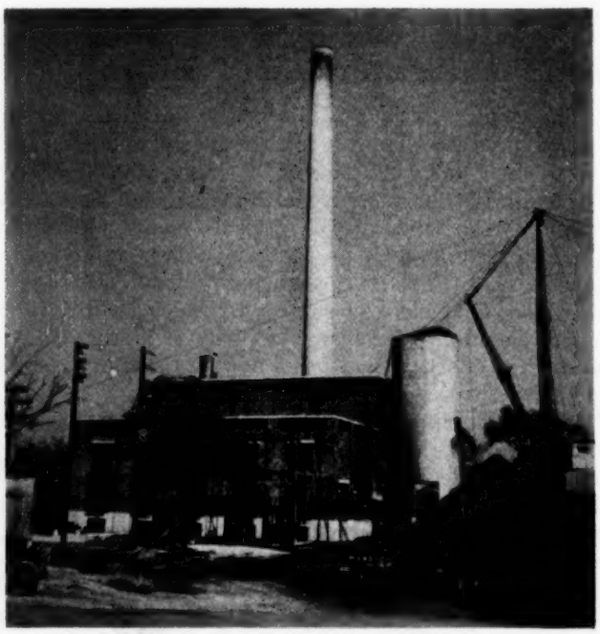

**Fig. 1—Exterior of plant showing fuel handling system**

field of slurry and the hill the opposite side. Slurry had accumulated in this pond to an average depth of probably twenty feet at the time of the installation and there was sufficient fuel to operate a power plant of the capacity contemplated for a number of years. However, the combined slurries from the two mines mentioned was much greater than necessary to operate the power plant. While these mines did not operate daily, their operation was sufficiently frequent that the slurry from the washing plants could be piled near the generating station and the remainder run off to the slurry field. Even if these mines should shut down indefinitely, at the end of a fifteen-year period there would be enough slurry accumulated in the slurry field to operate the generating station for another fifteen years.

### *Character of the Fuel*

The slurry from the New Hope mine is a much cleaner product than that from the Friar Tuck mine. This is due to the fact that in mining the coal from stripping pits invariably a certain percentage of clay is mixed in with the coal and this clay is discharged from the washing boxes mixed with the fine coal. The size of the slurry varies from approximately $^1/_{16}$ in. down to almost nothing, and the raw slurry, after the water has run out, usually contains approximately 24 per cent moisture and from 20 to 30 per cent ash, depending upon the location from which the coal was mined.

The clay mixed with the slurry is an exceedingly disagreeable element. In the first place, it dilutes the fuel and increases the ash content, and secondly, it was found that the clay held water and prevented this from running out of the piles and consequently lowered the heating value of the fuel. Also, it was not a constant quantity but varied from time to time through rather wide limits.

The slurry from the washing plants is discharged from a wooden trough into wooden settling tanks from which the water runs out and the coal settles in the tanks. There are two of these tanks. While one is being unloaded with a grab bucket, supported from a boom hoist, the other is being filled.

From the settling tanks the slurry is dipped out by the grab bucket and piled in piles around the base of the boom hoist as shown in Fig. 1. It was found on removing the slurry from the settling tanks that the moisture ran from 24 to 28 per cent, depending upon the clay content of the material, but that after four or five days of standing in piles the moisture was reduced to about 18 per cent. When it reached this point the material could be successfully handled on the traveling grate stokers.

For about two years the plant operated on this raw slurry, "sweetened" with about three per cent of $1^1/_4$-in. screenings. It was found at times that the screenings were necessary. This was true on rainy days when the moisture in the pile could not be reduced to 18 per cent, and also during exceedingly cold weather when the material in the piles froze solid and could not be handled. As previously stated, it was found that the clay, which varied considerably from different portions of the mine, decidedly affected the operation of the furnaces because water would not run out of the pile of material. With the development of washing equipment, last year a coal-washing table was installed which reduced the ash content from 28 per cent to about 14, and the moisture originally in the coal from about 24 per cent to about 14 per cent. The calorific value as fired varies from 8000 to 9000 Btu per lb. With this improvement in the quality of the fuel it has not been necessary up to the present time to use any other fuel in the operation of the plant.

### *Available Water Supply*

As in most coal mining locations, there was a scarcity of water and it was necesssary to use water which had accumulated from springs in an abandoned mine. This water, as will be seen from the accompanying analysis, was of an exceedingly disagreeable nature, and varied from 100 to 160 gr of solids per gallon. Its temperature was in the neighborhood of 60 F. It not only included sodium carbonate in large quantities, but appreciable amounts of calcium carbonate. Such water could not be used in boilers in a raw state for the presence of the sodium chloride and the sodium carbonates produced violent ebulitions.

WATER ANALYSIS

| | Gr per gal |
|---|---|
| Silica | 0.583 |
| Oxide of Iron and Alumina | 0.117 |
| Sodium Sulphate | 83.683 |
| Sodium Chloride | 0.379 |
| Sodium Carbonate (and Potassium Carb.) | 62.983 |
| Magnesium Carbonate | 3.110 |
| Calcium Carbonate | 4.665 |
| Total | 155.520 |

Accordingly evaporators were installed. These were of rather large capacity for the size of the plant and they have proved entirely successful.

The water also caused considerable trouble in the condensers. This was not anticipated for we rather thought that owing to the comparatively small quantity of calcium carbonate present that with reasonable temperatures in the spray pond there would be little trouble from incrustation on condenser tubes. This was an erroneous conclusion. The plant is located in exceedingly sultry and humid territory and the high temperature of the condensing water in the summer months resulted in considerable scale formation. This was aggravated by the growth of algae in the pond. These conditions, however, have been considerably mitigated by the use of chlorine treatment for the elimination of the algae, and by employing large quantities of condensing water.

### *Load Conditions*

The load initially was to consist of one strip mine, one shaft mine and the City of Linton, and it was contemplated that an additional strip mine would be added in the course of a few years. It was estimated that the three mine loads would ultimately produce a maximum demand of 2500 kw with swings as high as 3400 kw. The maximum demand from the mines would occur during the working period of seven hours, or at some time between 7 a.m. and 3 p.m. The maximum demand established by the city load would occur somewhere around six o'clock in the evening during the winter months. The city load was therefore essentially an

off-peak load, and the maximum demand would be in the neighborhood of 600 kw. The stripping shovel load was continuous 24 hr a day and was the most difficult load to handle. Initially, only one stripping shovel would be on the line. Later, however, two stripping shovels would operate. These stripping shovels are of the unbalanced Bucyrus-Erie type and of sixteen yards capacity. They are equipped with 800-hp synchronous motor-generator sets which supply d-c current for the control and hoist motors.

Fig. 2 is a chart showing the input of current to these shovels. The cycle is approximately of 60 sec duration. As the shovel digs into the bank of earth the load rises rapidly to a peak as high as 1600 kw and the digging load may exceed the lifting load. In Fig. 2 the lifting load is shown as reaching a peak of 1200 kw, but this curve was undoubtedly taken when the shovel was digging into soft earth. After the lift is made the earth is dumped and the dipper lowered. During the lowering process the motors operate on a regenerative cycle and therefore act as brakes, pumping current back into the line.

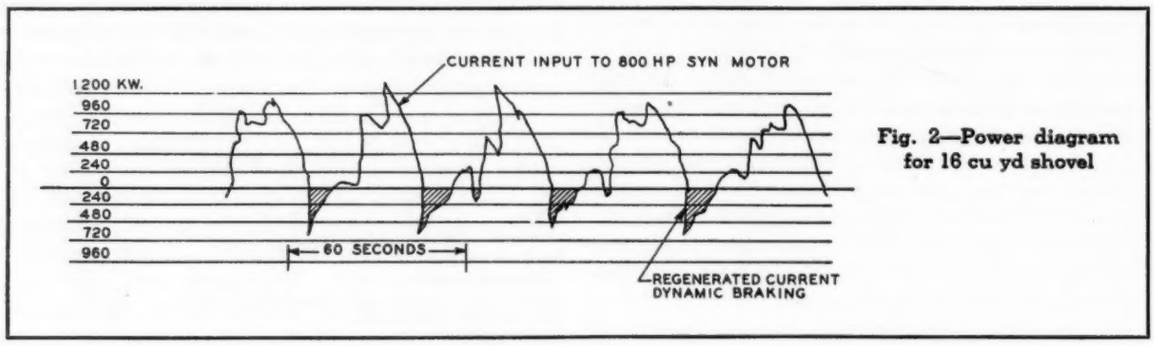


**Fig. 2—Power diagram for 16 cu yd shovel**

The speed of the turbine-generator supplying current to the shovel is consequently increased due to the load being entirely thrown off, and is further accelerated after the load has reached zero by the regenerative cycle pumping current back into the system. From the standpoint of regulation therefore the stripping shovel load is an exceedingly difficult load to handle. This load varies from a positive peak as high as 1600 kw to a negative peak as low as 700 kw. Due to the rapid fluctuations the governor alone cannot control the operation without light flicker so that equipment had to be selected with due consideration of the mass in the rotating parts. In the case of the Antioch plant, however, there was less light flicker in the City of Linton when operating on a 3000-kw turbine than there formerly was when public service current was used, generated by 20,000-kw units. This was due to the fact that our transmission lines were much shorter.

### *Plant Design*

Since the product used as fuel had little or no commercial value, the matter of efficiency in operation was of secondary consideration. It was therefore decided to employ used equipment as much as possible in the construction of the plant in order to keep down first costs. This was also desirable for the reason that old generating units were much heavier than new generating units and mass in the rotating parts was of the first importance in order to obtain proper regulation for the operation of lighting circuits from the transmission lines. As the plant must operate 24 hr per day, 365 days in the year, 100 per cent standby equipment was necessary in so far as generating capacity was concerned. As there were likely to be long periods of light loads at times, it was desirable to have at least one unit as small as 1000 kw. The plant was accordingly laid out with one 3000-kw main generating unit, one 2000-kw and one 1000-kw unit. With this arrangement one hundred per cent standby was provided in the turbine room and at the same time a very flexible arrangement of equipment, adapted to handle a widely fluctuating load condition.

Each turbine was equipped with a surface condenser, and the auxiliaries were arranged so that a large proportion of these could be operated either electrically or by steam to produce a heat balance suitable for varying load conditions.

### *Steam Generating Equipment*

For the steam generating equipment three 500-hp four-drum B&W Stirling boilers were used, designed for a working pressure of 225 lb per sq in. gage and equipped with superheaters for 100 deg superheat. These boilers were set with the mud drums 14 ft 3 in. above the boiler room floor and equipped with C. E. Coxe traveling grate stokers having a width of 10 ft $2^5/_8$ in. and an effective grate length of approximately 16 ft. The arrangement of boilers and stokers is shown in Fig. 3. A basement 12 ft in depth was provided under both the turbine room and the boiler room, and ashes are discharged from the ashpits to a steam jet conveyor which carries them to a point approximately one hundred feet from the building.

### *Coal Handling Equipment*

The coal handling equipment consists of a concrete silo 22 ft in diameter by approximately 65 ft in height, which delivers coal to a weigh larry operating on a overhead track in front of the boilers. The coal from the storage pile in front of the power house is lifted by means of a grab bucket from the storage pile into the silo, and from the silo it is delivered to the weigh larry.

Owing to the fineness of the material and the large percentage of moisture which tends to cause this to hang up in the bunkers, it was feared that there might be difficulty in this material descending from the silo. In order to obviate anything of this kind a sand conveyor, approximately 5 ft in width, was fitted into the bottom of the silo and adapted to deliver material from the silo to the weigh larry. It has been found, however, that it

is seldom necessary to use this conveyor, and the material descends considerably better than was anticipated.

A spray pond is located within approximately 300 ft of the plant for cooling the condensing water. The initial pond had a capacity to spray 6000 gal of water per minute.

### *Plant Extensions*

The plant was placed in operation in the fall of 1935, and by the summer of 1937 the company had placed the load of a number of mines on the plant which had previously not been contemplated. The result was that the peak load reached a maximum of 5200 kw and the spray pond was considerably overloaded. The boiler plant, however, proved sufficient to carry this load. Due to this overloaded condition, which left little reserve capacity in generating equipment for emergencies, and due to the fact that additional loads were contemplated, the 1000-kw unit originally installed was removed and a 5000-kw unit was installed in its place. The latter was placed in operation in January of this year and the plant now consists of one 3000-, one 2000- and one 5000-kw turbine-generator unit. The spray pond was enlarged to handle 12,000 gal of water per minute. With the installation of washing equipment for the slurry no screenings or other fuel is now required, and the firing conditions are as satisfactory as obtain in any well operated power installation. The success that attended the utilization of this fuel is indeed surprising.

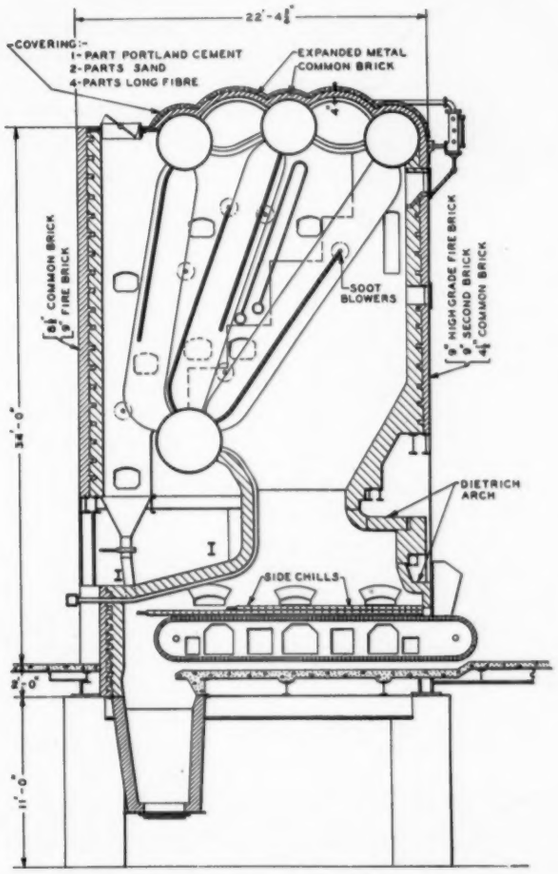


**Fig. 3—Section through steam generating unit**

Referring to Fig. 3, it will be noted that a water-cooled rear arch was installed over the traveling grate stoker, and the boilers have operated as high as 300 per cent rating. Repairs and replacements to the furnace and settings after three years of operation have been negligible, as have also the repairs on the stoker equipment, the furnace temperature seldom rising above 2000 F. All second-hand equipment used in the plant was rebuilt and reconditioned before installation, so that the maintenance charges on this portion of the plant are well within the limits of the maintenance charges on new equipment. As a matter of fact, fewer troubles developed in the starting up of this plant than usually occur where new equipment is purchased and many adjustments have to be made in the field.

### *Low Cost and Good Performance*

The total cost of the installation was approximately one-half the cost of a new installation, and the labor costs in operation are no greater and the maintenance costs no larger than would have occurred in a new installation. Since the fuel has little or no commercial value, the plant is a financial success.

The water rate on the plant varies considerably, but approximates 23 lb of steam per kilowatt-hour sent out. The evaporation averages 5.82 lb of water per pound of fuel, which corresponds to an efficiency of about 69 per cent for day in and day out operation. The installation demonstrates, first, the value of the traveling grate type of stoker for the burning of low grade fuels; second, that stripping shovel loads can be carried on small stations, providing the plant is properly designed to handle such loads; and, third, that essentially any kind of a water can be utilized when proper treatment has been worked out.

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## Within a Turbine

Steam enters a turbine at a temperature hot enough to set fire to a piece of wood and .03 of a second later leaves it at a temperature too cool for a comfortable bath. It may enter at a pressure of 1200 lb per sq in. and leave at the low vacuum of one-thirtieth of an atmosphere. The peripheral speed of recent large turbine rotors is approximately 820 mi per hr—80 mi faster than the speed of sound. These interesting facts were pointed out by W. E. Blowney of the General Electric Turbine Engineering Department in a paper presented recently to the Mid-West Power Conference at Chicago.

In 1903, the largest turbine-generator in service was a 5000-kw machine installed at Chicago. It was ten times larger than any previous unit. Today, the largest machine in service, also at Chicago, is forty-two times the size of that 1903 "colossal."

The initial steam temperature in turbines has reached the 950 F range; further progress to higher temperatures is limited until low creep materials are available. The study of creep, which is the permanent deformation of metals under stress while hot, is being intensively investigated, he said.

# Control for Sulzer Boiler

**In our April issue brief reference was made to a high-pressure extension, employing reciprocating engines, at the Emil Adolff works, Reutlingen (Germany). The following abstract from the translation of an article by C. Veit in *Zeitschrift des Vereines deutscher Ingenieure* describes in detail the automatic control of the Sulzer boiler and certain details of the high-pressure reciprocating feed pump.**

THE 1400-lb Sulzer boiler at this plant has a capacity of approximately 40,000 lb of steam per hr at 788 F total steam temperature and supplies two 1200-hp reciprocating engines which exhaust at varying back pressure, through oil separators, to heat exchangers that produce low-pressure steam for paper making. The oil removal from the condensate is accomplished by de-oiling means and by a coke filter in the feedwater line.

The Sulzer type of forced-circulation boiler is well known and will not be described here, but its control is of interest. For satisfactory operation it is essential that the coordination of heat supply to and heat delivery by the boiler be as nearly simultaneous as possible. Since the time for the flow of water and steam through the boiler varies with load changes, the foregoing requirement is met by two thermostatic elements responsive, respectively, to the final and an intermediate steam temperature. These are a part of the superheater circuit and are located outside the boiler setting in an easily accessible location. They are connected to the regulators operated by oil pressure for controlling the fuel, feedwater and superheater injection water. Fig. 1 shows the arrangement diagrammatically. In this *A* and *B* represent the boiler circuit and *C* and *D* superheater sections.

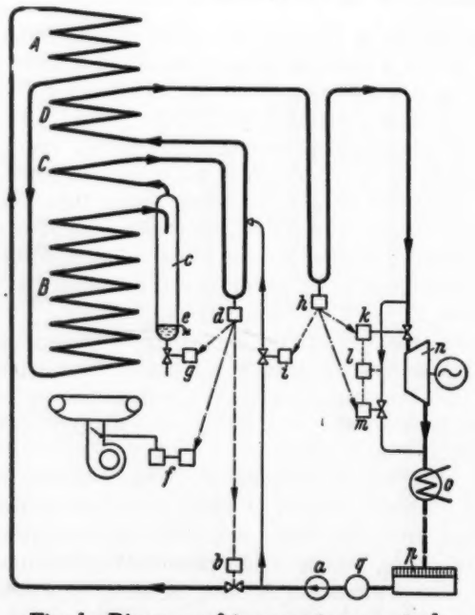


**Fig. 1—Diagram of temperature control**

Thermostat *d*, responding to the temperature at the beginning of superheat controls the fuel at *f* and the feedwater at *b;* also the water level within the vertical centrifugal steam separator *c* by means of the control *g*. There is a constant blowoff from separator *c*. When the water level in the separator rises due to too much feed, excess carryover into the superheater affects the temperature of the steam at the intermediate thermostat *d*. The latter acts to open the control valve *g* and thereby augments the constant blowoff and lowers the water level.

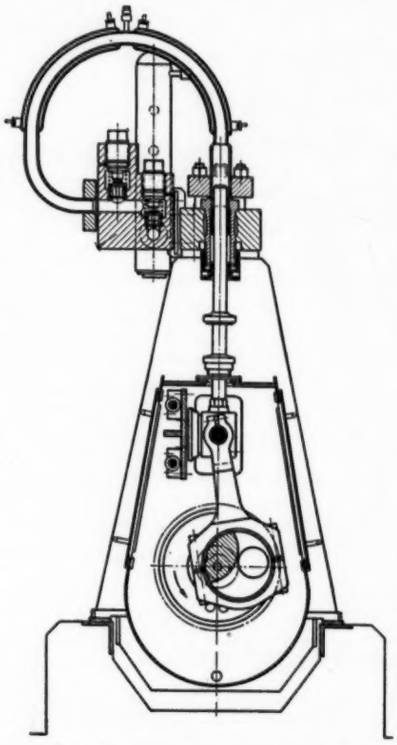

**Fig. 2—Section through high-pressure feed pump**

The intermediate thermostatic control is insufficient to maintain a uniform final steam temperature under rapid load changes, hence under such conditions the second thermostat *h*, which responds to the final steam temperature, is employed to control the feed injection into the superheater by means of control *i*. In addition to this, the regulators *k* and *m* protect the steam engines against too high or too low steam temperatures. These regulators are actuated by thermostat *h* at limiting temperatures beyond which the steam will bypass the engines.

### *High-Pressure Feed Pump*

The high-pressure feed pump has three single-acting plungers. In contrast to the usual pumps the suction and the pressure valves are not mounted on the pump cylinder but in an entirely separate valve box. The cylinder and the valve box are connected by a tube (see Fig. 2) the water within which serves, in effect, as an extension of the pump plunger. This avoids the direct contact of the hot feedwater with the plunger and packing. In order to avoid steam flashing in the pump suction, the feedwater is fed to the plunger pump by a centrifugal booster pump. The feed pump may deliver up to approximately 2100 lb per sq in.

Since January 1936 the boiler plant has operated satisfactorily for 13,600 working hours up until December 1937 when the article was written. Load fluctuations of from 30 to 40 per cent are met in 3 to 4 min without noticeable temperature and pressure changes.

# Load Distribution on Boilers

By Y. C. LU*

Shanghai Power Company, Shanghai, China

**The author, by a practical example, discusses the load distribution on boilers by the method of equal increments. The data used are hypothetical. In the example cited there are three groups of boilers of different capacities involving two different steam pressures. The analysis endeavors to prove that incremental input curves should be plotted on the basis of dollar input instead of Btu input which may show a false economy.**

IT is generally recognized that loading different units, either boilers, turbines or stations of a system, at their equal incremental input will result in a minimum energy input total and minimum fuel consumption; yet failure to realize certain points may lead to an erroneous conclusion, consequent waste in fuel and increase in operating cost. This article is an attempt to demonstrate the proper method of load distribution on boilers by a practical example, based on the following underlying principles, the units being placed on a truly comparable basis:

(*a*) Figures from efficiency tests are taken as the basis of comparison. For a plant having widely different units with different types of auxiliaries, the total auxiliary power for each unit is taken into consideration and the net efficiency curve is drawn.

(*b*) Input to boilers in terms of Btu in the fuel is a logical basis for a plant using similar grades of fuel. However, as the plant under consideration uses several kinds of coal of widely different compositions and prices, lowest Btu input may not mean the lowest dollar input which after all, is the ultimate aim. So in this case, input in terms of dollars is used.

(*c*) Boiler output in terms of pounds of steam per hour is a common standard if there be only one steam pressure in the station. For this particular case, having two or more steam pressures and consequently different turbine steam rates, equivalent boiler output in terms of kilowatts is used.[1]

Some boilers are banked after peak hours. The banking losses are not shown by incremental input curves and yet must be taken into account. Incremental input curves are supplemented by final input-output curves to show true economy.

Current literature on economical load distribution seems to emphasize only the incremental curves. Difficulty has been experienced in applying the incremental curves to the making of a loading schedule which can be readily understood by station operators. So it is attempted in this article to show the calculations which include the banking losses and furnish necessary data for a loading schedule. Also, on the loading schedule a system of designation is devised so that the station operators can understand at a glance the proper combination of units available to put into operation and the proper loading on each unit.

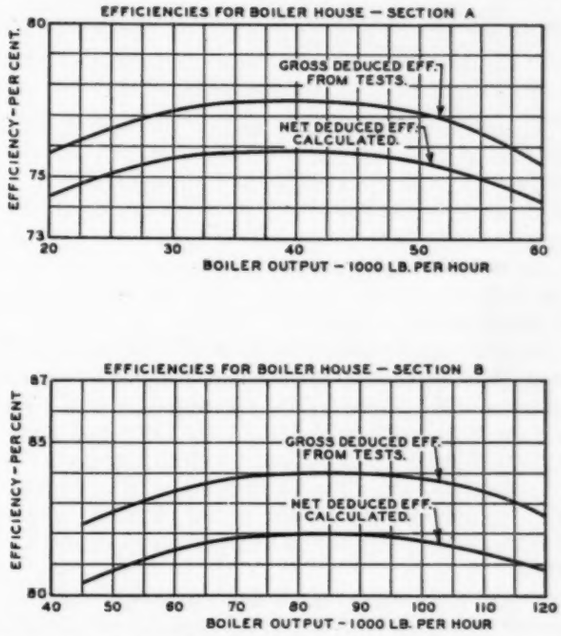


**Fig. 1—Gross deduced and net deduced efficiency curves, the upper set applying to section A and the lower to section B**

*The Problem*

The boiler plant considered in this study is divided into three sections. Section A consists of four boilers each having a maximum continuous capacity of 50,000 lb of steam per hour. Section B consists of ten boilers of 90,000 lb per hr each. Section C is the most recent and therefore the most efficient, and consists of four boilers of 210,000 lb per hr each. Units in Sections A and B are of 200 lb pressure and stoker fired. Those in Section C are of 400 lb pressure and fired by pulverized coal. Coal used in Section C is cheaper than that burned in A and B.

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[1] This particular case also provides for transferring steam from a higher to a lower pressure; the same boiler may have different equivalent outputs in terms of kilowatts, depending on the loading conditions. So the equivalent output in kilowatts is the most practical basis for comparison.

TABLE I. DATA FOR INCREMENTAL DOLLAR INPUT CURVES FOR SECTION C

Data: Gross deduced efficiency curve
Steam temperature curve
Auxiliary power curve
Coal price = $ 0.40/$10^6$ Btu

Standard conditions at superheater outlet: p = 375, t = 730 F
Standard feedwater temperature = 246 F
Total heat at p = 375, t = 730 = 1379.6 Btu/lb
Heat added = 1379.6 - (246 - 32) = 1165.6 Btu/lb or 11656 Btu/kwhr

Turbine steam rate = 10.00 lb/kwhr - 400-lb units) corresponding to std. conditions
12.00 lb/kwhr - 200-lb units)
Steam increases 5% in weight when transferred from 400-lb to 200-lb due to water added by desuperheater
Change in steam rate = 1% for deg 15 F change in temperature

| L = Load, 1000 lb/hr | 140 | 145 | 150 | 155 | 160 | 165 | 170 | 175 | 180 | 185 | 190 | 195 | 200 | 205 | 210 | 215 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| G D E = Gross deduced eff (from curve) % | 85.33 | 85.33 | 85.33 | 85.33 | 85.33 | 85.32 | 85.30 | 85.26 | 85.22 | 85.13 | 85.02 | 84.83 | 84.60 | 84.29 | 83.95 | 83.57 |
| $t_s$ = Steam Temp. F (from curve) | 703 | 708 | 714 | 718.5 | 723 | 727 | 731 | 734 | 738 | 741 | 744 | 747 | 750 | 753 | 756 | 758 |
| $Q_c$ = Change in steam rate = $\frac{.01}{15} \times 10\ (t_s - 730)$, (lb per kwhr) | + .1800 | + .1433 | + .1100 | + .0766 | + .0467 | + .0200 | - .0067 | - .0300 | - .0533 | - .0733 | - .0933 | - .1133 | - .1333 | - .1533 | - .1733 | - .1867 |
| Q = Turbine steam rate = 10 + $Q_c$, lb/kwhr | 10.180 | 10.143 | 10.110 | 10.077 | 10.047 | 10.020 | 9.9931 | 9.970 | 9.947 | 9.926 | 9.907 | 9.887 | 9.867 | 9.847 | 9.827 | 9.813 |
| K = Kw max. power (from curve) | 629 | 639 | 649 | 660 | 671 | 683 | 694 | 707 | 720 | 734 | 748 | 764 | 781 | 801 | 821 | 848 |
| $Q_E$ = Equiv. steam for max. = EQ, lb/hr | 6403 | 6481 | 6561 | 6651 | 6742 | 6844 | 6935 | 7049 | 7162 | 7286 | 7410 | 7554 | 7706 | 7887 | 8088 | 8321 |
| R = Eff. ratio = $\frac{L}{(L + Q_E)}$ | .95626 | .95722 | .95809 | .95886 | .95957 | .96017 | .96080 | .96128 | .96173 | .96211 | .96246 | .96270 | .96290 | .96295 | .96291 | .96274 |
| N D E = Net deduced eff. = G D E x R, % | 81.598 | 81.679 | 81.754 | 81.819 | 81.880 | 81.922 | 81.956 | 81.959 | 81.959 | 81.904 | 81.828 | 81.666 | 81.461 | 81.167 | 80.836 | 80.456 |
| h = Total heat at p = 375 and $t_s$, Btu/lb | 1364.2 | 1367.3 | 1370.0 | 1372.6 | 1375.2 | 1377.5 | 1380.0 | 1381.9 | 1383.6 | 1385.4 | 1386.4 | 1388.4 | 1390.4 | 1392.2 | 1394.0 | 1395.0 |
| $h_a$ = Heat added = h - (246 - 32) = h - 214, Btu/lb | 1150.2 | 1153.3 | 1156.0 | 1158.6 | 1161.2 | 1163.5 | 1166.0 | 1167.9 | 1169.6 | 1171.4 | 1172.4 | 1174.4 | 1176.4 | 1178.2 | 1180.0 | 1181.0 |
| $H_a$ = Total heat added = Q $h_a$, Btu/lb | 11709 | 11699 | 11687 | 11677 | 11667 | 11658 | 11652 | 11644 | 11636 | 11628 | 11615 | 11611 | 11608 | 11602 | 11596 | 11589 |
| E N D E = Equiv. net deduced eff. = NDE x $\frac{11656}{H_a}$, % | 81.229 | 81.386 | 81.537 | 81.672 | 81.803 | 81.908 | 81.984 | 82.043 | 82.100 | 82.101 | 82.117 | 81.983 | 81.797 | 81.545 | 81.254 | 80.921 |
| H = Grand total heat added = $\frac{L \times 1165.6}{NDE \times 10^6}$, (million Btu per hr) | 200.894 | 207.667 | 214.430 | 221.212 | 227.961 | 234.804 | 241.695 | 248.626 | 255.552 | 262.647 | 269.693 | 277.244 | 284.998 | 293.026 | 301.248 | 309.690 |
| C = Dollar input per hr = 0.40 x H | 80.3576 | 83.0668 | 85.7720 | 88.4848 | 91.1924 | 93.9216 | 96.6780 | 99.4504 | 102.2208 | 105.0588 | 107.8772 | 110.8978 | 113.9992 | 117.2104 | 120.4992 | 123.8760 |
| $I_{(10)}$ = Incremental dollar input per 1000 kwhr when steam is used in 400-lb turbines = $\frac{C_3 - C_1}{10} \times 10 = C_3 - C_1$ (Increment at $C_2$) | | 5.4144 | 5.4188 | 5.4204 | 5.4368 | 5.4656 | 5.5288 | 5.5428 | 5.6084 | 5.6564 | 5.8390 | 6.1220 | 6.3126 | 6.5000 | 6.6656 | |
| $I_{(12)}$ = Incremental dollar input per 1000 kwhr when steam is used in 200-lb turbines = $\frac{C_3 - C_1}{10} \times \frac{12}{1.05}$ (Increment at $C_2$) | | 6.1875 | 6.1926 | 6.1944 | 6.2131 | 6.2461 | 6.3183 | 6.3943 | 6.4093 | 6.4641 | 6.6728 | 6.9962 | 7.2140 | 7.4282 | 7.6174 | |

The steam load is carried mainly by Sections B and C. Besides those steaming there are always one or two units in Sections A and B banked for emergency in case one of the units in Section C drops off during the peak. Approximately half of the station load is carried on the 400-lb turbines which are of the two-cylinder, 3000-rpm type and show better economy than the older 1500-rpm machines in the station.

The total rated capacity of the 400-lb turbines is 66,000 kw and that of the 200-lb turbines is 120,000 kw making a total of 186,000 kw. The steam rate for the 400-lb turbines is approximately 10 lb per kwhr and that for 200-lb turbines is 12 lb per kwhr. Both turbine capacity and steam rate are different in summer and winter, the foregoing values being for the winter months. Separate calculations are required in order to cover summer conditions.

The total steam required by the 400-lb turbines is 660,000 lb per hr and the total capacity of the 400-lb boilers is 840,000 lb per hr, thus leaving a maximum margin of 180,000 lb per hr which may be transferred to to the 200-lb machines. The reducing valve and desuperheater are ample in size so that a large amount of steam can be transferred when some of the 400-lb machines are out of commission.

The problem of load distribution on the boilers is analyzed as follows:

(1) During the peak period when there is a large demand for steam how much 400-lb steam should be transferred to the 200-lb station?

(2) As the station load is decreasing from the peak, should the load on Section C boilers be lowered and the transfer shut off, or should the load on Sections A and B be lowered and the transfer left as it was at the peak?

(3) As the station load is still decreasing, should all boilers in Sections A and B be cut down below their most efficient point or should some boilers be kept at their most efficient steaming rate and the remainder of the boilers banked?

### *Incremental Input Curves*

Figs. 1 and 2 show the gross deduced efficiency curves of the three sections mentioned. The efficiency is the difference between unity and the sum of all the losses obtained from tests. Then the net deduced efficiency curves are drawn by taking auxiliary power into con-

TABLE II. DATA FOR INPUT OUTPUT CURVES FOR DIFFERENT BOILER LOADINGS
(complete figures omitted to save space)

Data:

Loading at equal incremental dollar input:

| Section B | Section C | NDE (Sec C) | from curve, % | Dollar Input = $\frac{L \times 1165.6}{NDE} \times \frac{40}{10^6}$ (Sec C) |
|---|---|---|---|---|
| 90,000 lb/hr | - 204,000 lb/hr | 81.590 | | 116.874 |
| 80,000 " | - 201,500 " | 81.725 | | 114.955 |
| 70,000 " | - 199,500 " | 81.610 | | 113.686 |
| 60,000 " | - 198,500 " | 81.890 | | 113.071 |

Banking losses (test results):
8 hr banking for loading (B.L.) (11 x $10^6$ Btu/boiler hr or 11 x .42 = $4.62/boiler hr (Section B)
( 4 x $10^6$ Btu/boiler hr or 4 x .42 = $1.68/boiler hr (Section A)
24 hr banking for standby (B.S.) ( 5 x $10^6$ Btu/boiler hr or 5 x .42 = $2.10/boiler hr (Section B)
( 2 x $10^6$ Btu/boiler hr or 2 x .42 = $0.84/boiler hr (Section A)

Maximum continuous capacity:
210,000 lb/hr (Section C)
90,000 lb/hr (Section B)
80,000 lb/hr (Section A)
Transfer to 200-lb TG = 1.05 x transfer from 400-lb TG

| Boiler Loading, 1000 lb/hr Sec C | Sec B Loaded | Sec B Banked | Sec A Loaded | Sec A Banked | Dollar Input Sec C Individual | Sec C Group | Sec B Individual | Sec B Group | Sec B Banking | Sec A Individual | Sec A Group | Sec A Banking | Steam, 1000 lb/hr Total | 400-lb | 200-lb | Trans. from 400-lb | Trans. to 200-lb + Trans. | Equiv. Kwhr 400-lb TG | 200-lb TG | Total | $/equiv. kwhr |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 4 | 8 | 1 | 2 | 1 | | | | | | | | | | | | | | | | | |
| 210 | 90 | B.S. | 80 | B.S. | 120.499 | 481.996 | 84.634 | 437.072 | 2.100 | 33.007 | 66.014 | 0.840 | 988.082 | 840 | 880 | 180 | 189 | 1009 | 66,000 | 84,083 | 150,083 | .0065832 |
| " | " | " | 2 40 | " | | " | | " | " | 26.293 | 52.586 | " | 978.894 | " | 800 | " | " | 989 | " | 82,417 | 148,417 | .0065665 |
| " | " | " | 2 30 | " | | " | | " | " | 19.788 | 39.566 | " | 981.874 | " | 790 | " | " | 989 | " | 80,790 | 146,790 | .0065594 |
| 4 | 8 | 1 | 1 | 1 B.S. | | | | | | | | | | | | | | | | | |
| 210 | 90 | B.S. | 80 | 1 B.L. | | 481.996 | | 437.072 | 2.100 | 33.007 | 2.520 | 956.695 | 840 | 770 | 180 | 189 | 959 | 66,000 | 79,917 | 145,917 | .0065864 |
| " | " | " | 1 40 | " | | " | | " | " | 26.293 | " | 949.981 | " | 760 | " | " | 948 | " | 79,083 | 145,083 | .0065478 |
| " | " | " | 1 30 | " | | " | | " | " | 19.733 | " | 943.471 | " | 750 | " | " | 938 | " | 78,250 | 144,250 | .0062405 |
| 4 | 8 | 1 | 1 B.S. | | | | | | | | | | | | | | | | | | |
| 210 | 90 | B.S. | - | 2 B.L. | 120.499 | 481.996 | 84.634 | 437.072 | 2.100 | - | 4.200 | 935.368 | 840 | 720 | 180 | 189 | 909 | 66,000 | 75,750 | 141,750 | .0063781 |
| 4 | 8 | | | | | | | | | | | | | | | | | | | | |
| 204 | 90 | " | - | " | 116.874 | 466.896 | | 427.072 | " | - | " | 909.662 | 816 | 720 | 156 | 163.8 | 883.8 | " | 73,650 | 139,650 | .0063139 |
| 4 | 8 | | | | | | | | | | | | | | | | | | | | |
| 201.6 | 90 | " | - | " | 114.955 | 459.820 | 48.372 | 388.896 | " | - | " | 894.696 | 806 | 640 | 146 | 153.3 | 793.3 | " | 66,108 | 132,108 | .0064896 |
| 4 | 8 | | | | | | | | | | | | | | | | | | | | |
| 199.5 | 70 | " | - | " | 113.686 | 454.784 | 42.535 | 340.280 | " | - | " | 881.284 | 798 | 560 | 138 | 144.9 | 704.9 | " | 58,742 | 124,742 | .0064341 |
| 4 | 8 | | | | | | | | | | | | | | | | | | | | |
| 199.5 | 60 | " | - | " | 113.071 | 452.284 | 36.614 | 292.912 | " | - | " | 731.496 | 794 | 480 | 134 | 140.7 | 620.7 | " | 51,725 | 117,725 | .0064233 |
| 4 | 8 | | | | | | | | | | | | | | | | | | | | |
| 165 | 60 | " | - | " | 93.932 | 375.888 | | 292.912 | " | - | " | 674.900 | 660 | 480 | 0 | 0 | 480 | " | 40,000 | 106,000 | .0062970 |
| 4 | 8 | | | | | | | | | | | | | | | | | | | | |
| 160 | 60 | " | - | " | 91.192 | 364.768 | | 292.912 | " | - | " | 657.680 | 640 | 480 | 0 | 0 | 480 | 84,000 | 40,000 | 104,000 | .0062844 |

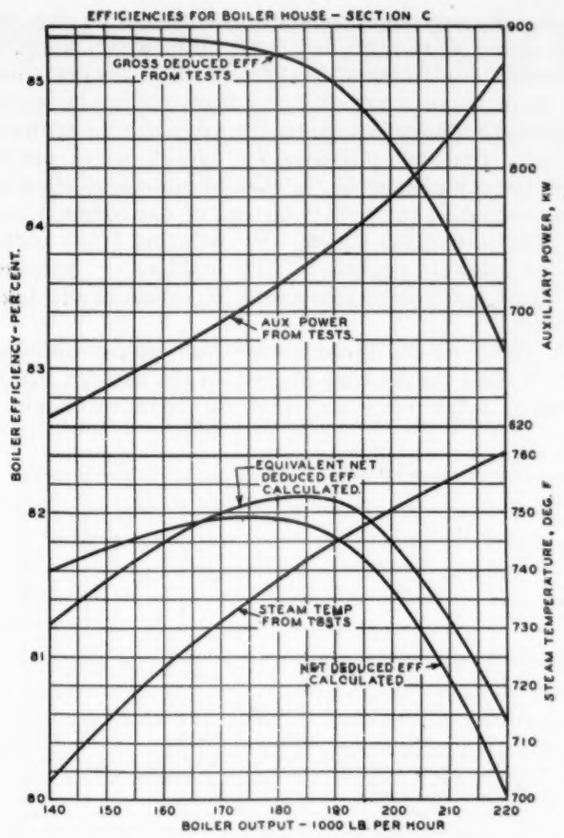

EFFICIENCIES FOR BOILER HOUSE – SECTION C
GROSS DEDUCED EFF FROM TESTS
AUX POWER FROM TESTS
EQUIVALENT NET DEDUCED EFF CALCULATED
STEAM TEMP FROM TESTS
NET DEDUCED EFF CALCULATED
BOILER EFFICIENCY – PER CENT.
AUXILIARY POWER, KW
STEAM TEMPERATURE, DEG. F
BOILER OUTPUT – 1000 LB. PER HOUR

**Fig. 2—Efficiency curves for section C**

sideration. Data on auxiliary power are taken at the same time as the boiler tests. Calculations are shown on the accompanying tables.

Next the *equivalent* net deduced efficiency curves are drawn. By equivalent net deduced efficiency is meant the efficiency of a boiler operating at standard steam temperature which would give the same overall plant efficiency as the actual boiler operating at actual efficiency and temperature. By these curves the most economical loading on the boilers can be determined without considering steam temperature and its effect on turbine operation. Also, when the turbine steam rate is considered, no correction for temperature is necessary as it is already included in the boiler data. The equivalent efficiencies are obtained by ratio of turbine steam rates and heat added per pound of steam, respectively, at the operating and standard temperatures. Calculations for Section C are shown on Table I. For simplicity the net deduced efficiency is taken to be the equivalent efficiency for Sections A and B; calculations for Sections A and B are similar but are not here reproduced. Table II gives data for plotting input-output curves for different boiler loadings. In order to save space the complete table is not here reproduced, but only enough to show the procedure involved in computing these data for the curves.

The difficulty in obtaining the incremental input curves, Fig. 3, is greater than first appears. The increment is so small compared with the total dollar input figures that slide-rule calculations are not sufficient to make a curve of reasonable accuracy. Even with four decimal places worked out by a calculating machine some points are still off the smooth curve. This will be apparent by reference to the two lower curves shown in Fig. 3.

### *Input-Output Curves*

Once the incremental input curves are obtained there will be no difficulty in loading boilers during higher output of the station as it is well known that loading these units at their equal incremental inputs will result in a minimum energy input total.

This general principle must be modified somewhat in order to meet practical conditions. Referring to the loading schedule or the input-output curves, Fig. 4, every boiler operating has to be loaded to its maximum

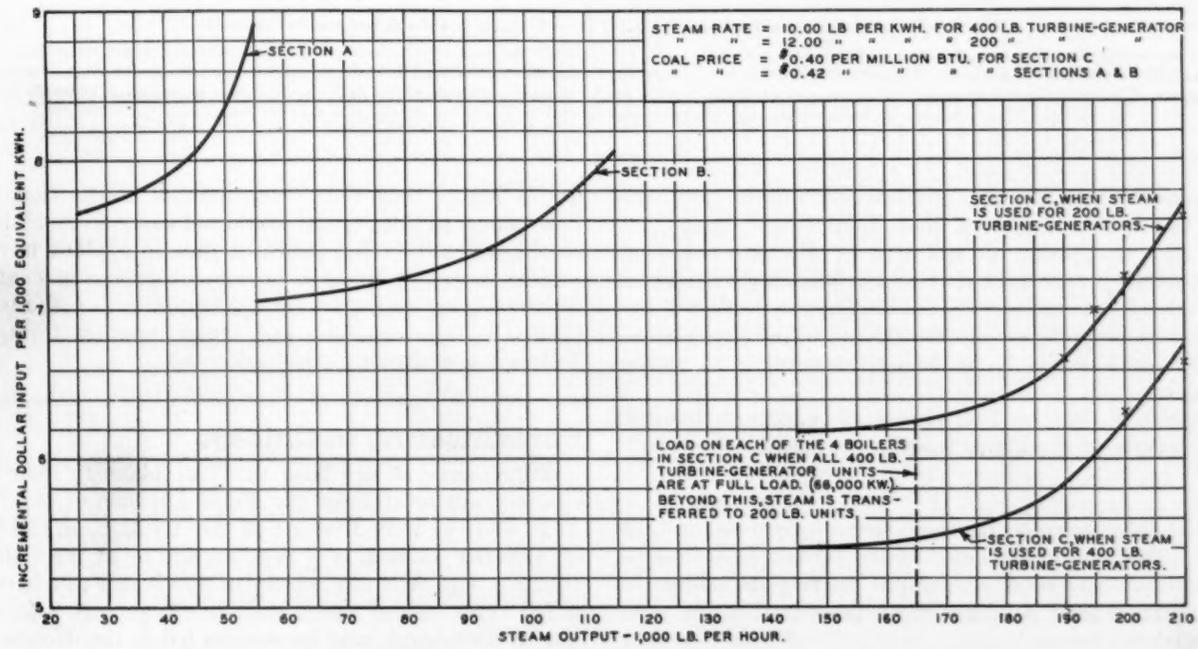

STEAM RATE = 10.00 LB PER KWH. FOR 400 LB. TURBINE-GENERATOR
" " = 12.00 " " " " 200 " " "
COAL PRICE = $0.40 PER MILLION BTU. FOR SECTION C
" " = $0.42 " " " " SECTIONS A & B
SECTION A
SECTION B.
SECTION C, WHEN STEAM IS USED FOR 200 LB. TURBINE-GENERATORS.
LOAD ON EACH OF THE 4 BOILERS IN SECTION C WHEN ALL 400 LB. TURBINE-GENERATOR UNITS ARE AT FULL LOAD. (68,000 KW.) BEYOND THIS, STEAM IS TRANSFERRED TO 200 LB. UNITS.
SECTION C, WHEN STEAM IS USED FOR 400 LB. TURBINE-GENERATORS.
INCREMENTAL DOLLAR INPUT PER 1,000 EQUIVALENT KWH.
STEAM OUTPUT – 1,000 LB. PER HOUR.

**Fig. 3—Incremental dollar input curves for boilers, based on calculations in tables**

continuous capacity to meet the 150,000 kw station peak, regardless of individual boiler efficiencies and inputs. As the station load decreases from the peak, load on Section A being at the highest incremental inputs, is decreased first. After all the boilers in Section A have been banked, loading on Sections C and B is decreased corresponding to equal incremental inputs. At 118,000 kw the load on Section B is at the minimum without banking boilers. If the loading at equal increments is adhered to, one boiler in Section B must be banked. The input-output curves show that the gain at equal increments cannot offset the banking loss. All boilers are therefore kept steaming, and loading on Section C is decreased until the "transfer" is shut off at 165,000 lb per hr (see Fig. 3). This particular case indicates that repairs the maximum total steam output from this section would be 630,000 lb per hr, all of which would be consumed by the three 400-lb turbines. The incremental input curves show that the three boilers in Section C should be loaded always at their maximum continuous rating. This case indicates the importance of putting the incremental input on the basis of equivalent kilowatt-hour output of the boilers instead of the common basis in 1000 lb of steam per hr. By reducing boiler output to the kilowatt equivalent, the problem of "transfer" of steam, as is here illustrated, is automatically taken care of.

A glance at the incremental input curves will show that if these curves were plotted on the basis of Btu instead of dollar input, the curve for Section C would be

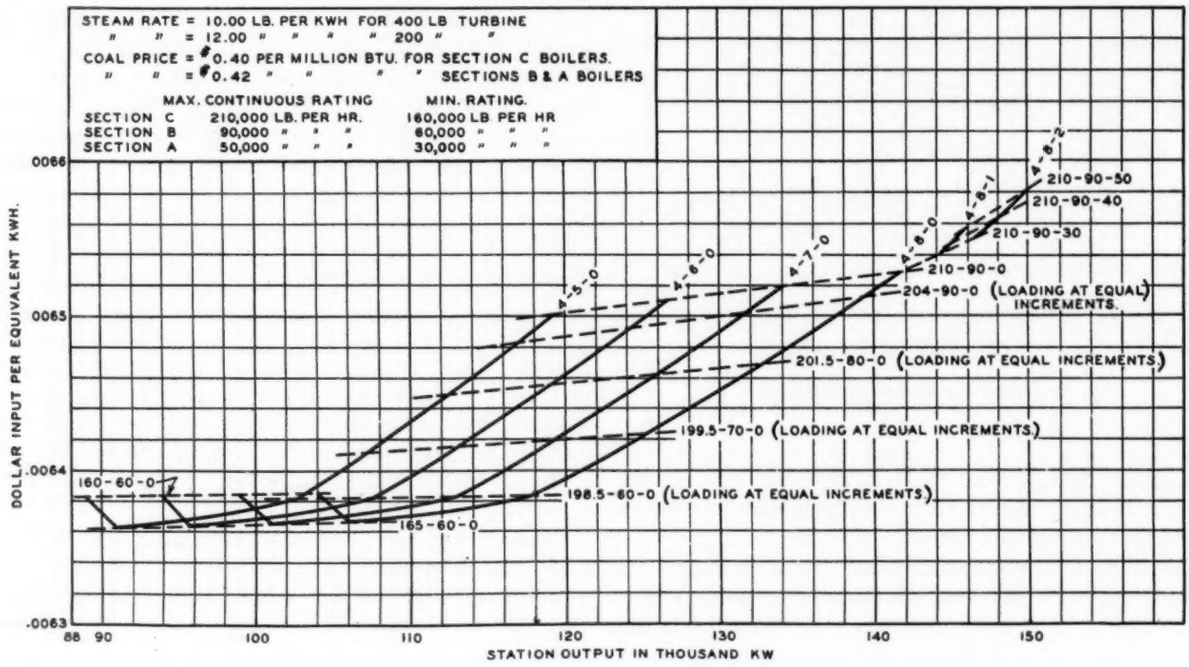

STEAM RATE = 10.00 LB. PER KWH FOR 400 LB TURBINE
" " = 12.00 " " " " 200 " "
COAL PRICE = $0.40 PER MILLION BTU. FOR SECTION C BOILERS.
" " = $0.42 " " " " SECTIONS B & A BOILERS
MAX. CONTINUOUS RATING
MIN. RATING.
SECTION C 210,000 LB. PER HR. 160,000 LB PER HR
SECTION B 90,000 " " " 60,000 " " "
SECTION A 50,000 " " " 30,000 " " "
DOLLAR INPUT PER EQUIVALENT KWH.
.0066
.0065
.0064
.0063
88 90 100 110 120 130 140 150
STATION OUTPUT IN THOUSAND KW
4-5-0
4-6-0
4-7-0
4-8-0
4-8-1
4-8-2
210-90-50
210-90-40
210-90-30
210-90-0
204-90-0 (LOADING AT EQUAL INCREMENTS.
201.5-80-0 (LOADING AT EQUAL INCREMENTS)
199.5-70-0 (LOADING AT EQUAL INCREMENTS)
198.5-60-0 (LOADING AT EQUAL INCREMENTS)
160-60-0
165-60-0

**Fig. 4—Boiler loading schedule showing input-output values for winter months**

The combinations of figures, 4–5–0, 4–6–0, 4–7–0, 4–8–0, 4–8–1 and 4–8–2 signify the number of boilers operating; for instance, 4–8–1 means that 4 boilers are operating in section C, 8 in section B and 1 in section A. The remaining combinations indicate the loadings on individual boilers; for instance, 210–90–30 signifies that section C units are carrying 210,000 lb per hr, those in section B 90,000 lb per hr and those in section A 30,000 lb per hr. Besides the boilers operating there is 1 boiler in section B and 1 in section A banked for standby

loading boilers at equal incremental inputs alone will not result in a minimum input total.

Load on Section C can still be decreased beyond 165,000 lb per hr but this would require higher input as shown by the curves. So the loading on Section C and B is kept, respectively, at 165,000 and 60,000 lb per hr and boilers in Section B are banked according to station demand.

When the station load rises from minimum to peak, the reverse of the above loading is applied.

### *Conclusion*

The input-output curves answer all questions in loading. The graph of these curves serves as a loading schedule. The most economical loading on boilers for any station load is indicated by the curve at the minimum dollar input.

In case one boiler in Section C is out for overhaul or drawn higher than it is shown, which means that the four boilers in this section would not carry so much load as they should. This indicates that in a plant having various types of firing arrangement requiring coal of different composition and cost, comparison on the basis of Btu is false economy and boiler input in terms of dollars is the ultimate standard.

## Fuels Meeting in Chicago

Announcement has been made of plans to hold the second joint meeting of the Fuels Division of the A.S.M.E. and the Coal Division of the A.I.M.E. in Chicago on October 13 and 14. Sessions will be at the Palmer House. The Western Society of Engineers will participate. The first of these joint meetings was held last fall in Pittsburgh and its success led to the decision to make it an annual event.

# INSULATION COMPUTATIONS

By F. PEITER and DR. M. J. FISH

Engineering Dept., Combustion Engineering Company, Inc.

**The authors present a method of computation for determining the insulation required for vertical walls of boilers. The formulas are developed and explained, and typical examples are included to illustrate the use of the formulas.**

THE laws of heat insulation, together with much research data, have been known for many years. However, with the vast development in steam generating equipment and the accompanying high temperatures, it has become necessary to use the known data to the greatest possible advantage in the design of steam generating units. The authors here present a method of computation for vertical walls in boilers which they have used successfully for many years.

The general method will be developed and then a few typical problems will be presented to show how the equations are used. Fig. 1 indicates how the conductivity of the particular insulating material varies with the mean value of the temperature through the material. The relation differs for each material but for most commercial insulations can be represented by a straight line, the equation of which is

$$c = mt + b \qquad (1)$$

where,

$c$ = the conductivity of the material in Btu per sq ft of surface, per degree Fahrenheit, per inch of thickness,

$m$ = the slope of the straight line or the tangent of angle $\phi$,

$t$ = the mean temperature of the material in degrees Fahrenheit,

$b$ = the value of $c$ when $t$ is zero.

The data from many exhaustive tests are available to draw such curves for all commercial insulations. Manufacturers furnish these data in their catalogs.

In Fig. 2, let

$T_1$ = the temperature of the hot surface in degrees Fahrenheit,

$T_2$ = the temperature of the cold surface in degrees Fahrenheit,

$t$ = the temperature of a point in the wall at a distance $x$ from the cold surface in degrees Fahrenheit.

$H$ is the rate of heat transfer from the cold surface of the material at temperature $T_1$ in Btu per sq ft of surface per hour.

Since the flow of heat through the material is continuous, in a given time, the same quantity of heat flows through the wall by conduction as flows off the surface into the surrounding air by radiation and convection.

It is well known that the quantity of heat flowing across an infinitesimal length $dx$ (see Fig. 2) can be expressed as

$$H = c\frac{dt}{dx} \qquad (2)$$

where $dt$ is the temperature change in the length $dx$.

Replacing $c$ in (2) by its value $mt + b$ from equation (1) and integrating the result across the entire cross-section of the wall, we have

$$\int_0^L H dx = \int_{T_2}^{T_1} (mt + b)dt$$

which becomes

$$HL = \frac{m}{2}(T_1^2 - T_2^2) + b(T_1 - T_2).$$

By adding $\frac{b^2}{m^2}$ to both sides of the latter equation, it can be rewritten as

$$\left(T_1 + \frac{b}{m}\right)^2 = \frac{2H}{m}\left[L + \frac{(b + mT_2)^2}{2Hm}\right] \qquad (3)$$

If $T_1 + \frac{b}{m}$ be denoted by $T_n$ and $L + \frac{(b + mT_2)^2}{2Hm}$ by $L_n$, equation (3) can be written as $T_n^2 = \frac{2H}{m} L_n$ which is recognized as a parabola, both of whose coordinate axes have undergone a parallel displacement. The amount of displacements of the new axes $O_nT_n$ and $O_nL_n$ from the old axes $OT_1$ and $OL$ is given by $\frac{b}{m}$ and $\frac{(b + mT_2)^2}{2Hm}$, respectively.

Fig. 3 shows the curve of equation (3) referred to both the new and the old axes and superimposed on the cross-section of the wall. The heavy portion of the curve is the temperature gradient through the wall. This is drawn more accurately than if it is assumed to be a straight line gradient. The mean temperature is not at the center of the cross-section but is displaced to the cold side. The thicker the wall, the more pronounced is this deviation.

Values of the emissivity, $H$, are found from a modification of the Stefan-Boltzmann Law that has been suggested by Professor W. Trinks.[1] This is,

$$H = 0.155\left[\left(\frac{T+460}{100}\right)^4 - \left(\frac{T_a+460}{100}\right)^4\right] + 0.25\,(T-T_a)^{1/4} \qquad (4)$$

where $H$ has the same meaning as before, $T$ = the surface temperature of the wall in degrees Fahrenheit and $T_a$ = the temperature of still ambient air in degrees Fahrenheit.

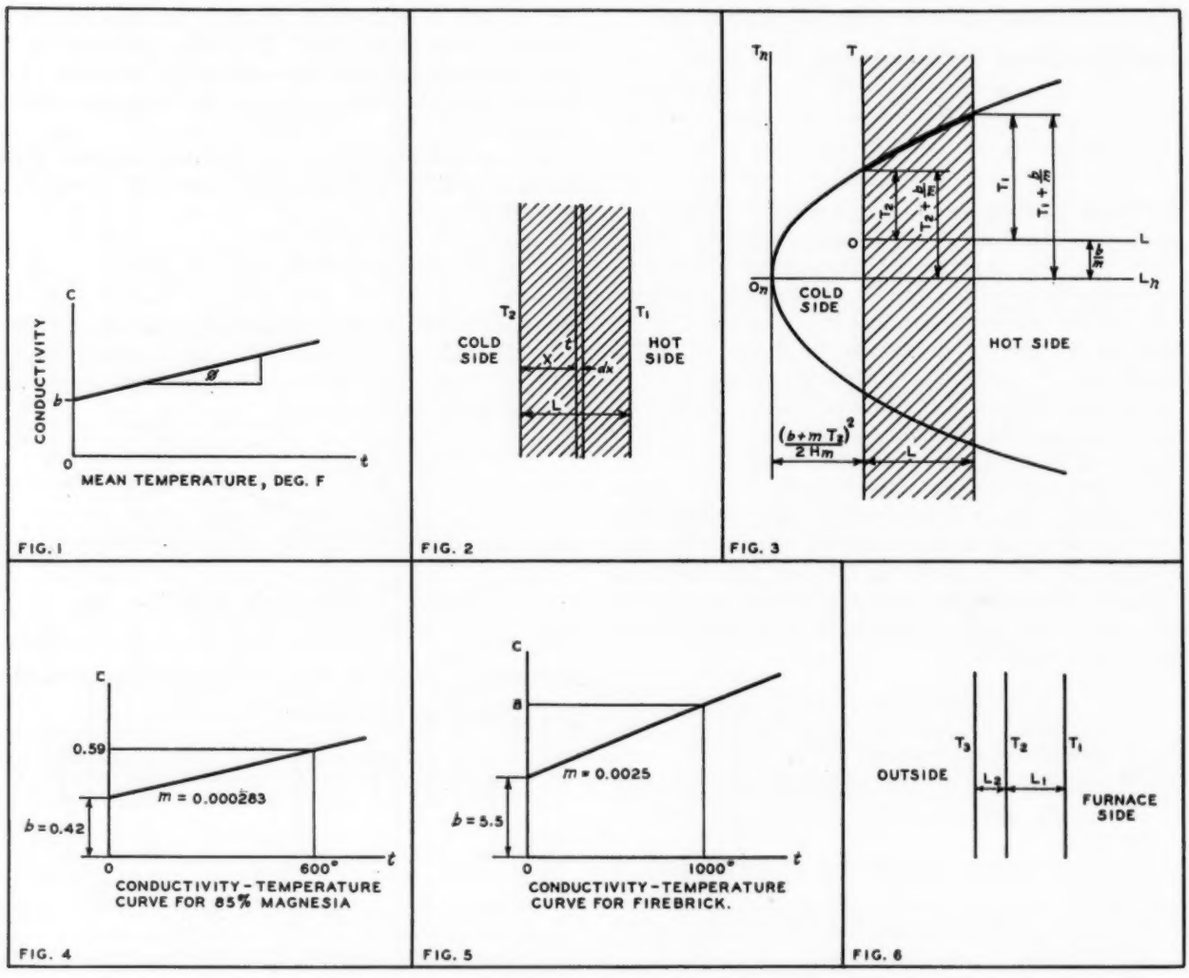
The first term of equation (4) represents the surface loss due to radiation while the second term is the "convection" loss from the surface.

$$\left(T_1+\frac{b}{m}\right)^2 = \frac{2HL}{m} + \left(\frac{b}{m}+T_2\right)^2$$

Solving for $L$, $L = \frac{m}{2H}\left[\left(T_1+\frac{b}{m}\right)^2 - \left(T_2+\frac{b}{m}\right)^2\right]$

Since $\left(T_1+\frac{b}{m}\right)^2 - \left(T_2+\frac{b}{m}\right)^2 = \left(T_1+\frac{b}{m}+T_2+\frac{b}{m}\right)\left(T_1+\frac{b}{m}-T_2-\frac{b}{m}\right)$

$$L = \frac{m}{2H}\left(T_1+T_2+\frac{2b}{m}\right)(T_1-T_2) \qquad (5)$$




**Figs. 1 to 6—Illustrating basis of formulas**

In the accompanying Table are given the values of $H$ for values of $T_2$ ranging from 100 to 400 F with still ambient air at 70, 80, 90 and 100 F. The figures in parenthesis in each column represent the approximate value of $H$ for one degree temperature difference to make interpolation easy.

The problem which is usually encountered is to find the thickness of insulation for given temperatures of the hot and cold surfaces. Equation (3) can be written in a more convenient form for this purpose as follows:

[1] *Industrial Furnaces* by W. Trinks, vol. 1, 1923 edition, John Wiley and Sons, Inc.

*Examples*

The following examples illustrate applications of the foregoing discussion and formulas.

EXAMPLE 1. Given a 2-in. thick wall of 85 per cent magnesia as an insulating medium with a cold surface temperature of 120 F and ambient air at 70 F, the problem is to find the temperature of the hot surface.

The slope of the conductivity curve (Fig. 4), $m = \frac{0.59-0.42}{600} = 0.000283$

From the Table, $H$ = 86 Btu.

Equation (3), $\left(T_1 + \frac{b}{m}\right)^2 = \frac{2H}{m}\left[L + \frac{(b + mT_2)^2}{2Hm}\right]$

becomes

$$T_1 = \left\{\frac{2 \times 86}{0.000283}\left[2 + \frac{(0.42 + 0.000283 \times 120)^2}{2 \times 86 \times 0.000283}\right]\right\}^{1/2} - \frac{0.42}{0.000283}$$

$= 1960 - 1484 = 476$ F, temperature of the hot surface.

EXAMPLE 2. Given a hot surface temperature of 1950 F and a cold surface temperature of 380 F, air at 120 F with ambient air again at 70 F. The problem is to find $T_2$ and $T_1$.

From the Table, $H = 86$ Btu. To obtain $T_2$, use equation (3) and refer to Fig. 4.

$$\left(T_2 + \frac{b_2}{m_2}\right)^2 = \frac{2H}{m_2}\left[L_2 + \frac{(b_2 + m_2 T_3)^2}{2Hm_2}\right] \qquad (3)$$

$$T_2 = \left\{\frac{2 \times 86}{0.000283}\left[2 + \frac{(0.42 + 0.000283 \times 120)^2}{2 \times 86 \times 0.000283}\right]\right\}^{1/2} - \frac{0.42}{0.000283}$$

$= 1960 - 1484 = 476$ F, as in Example 1.

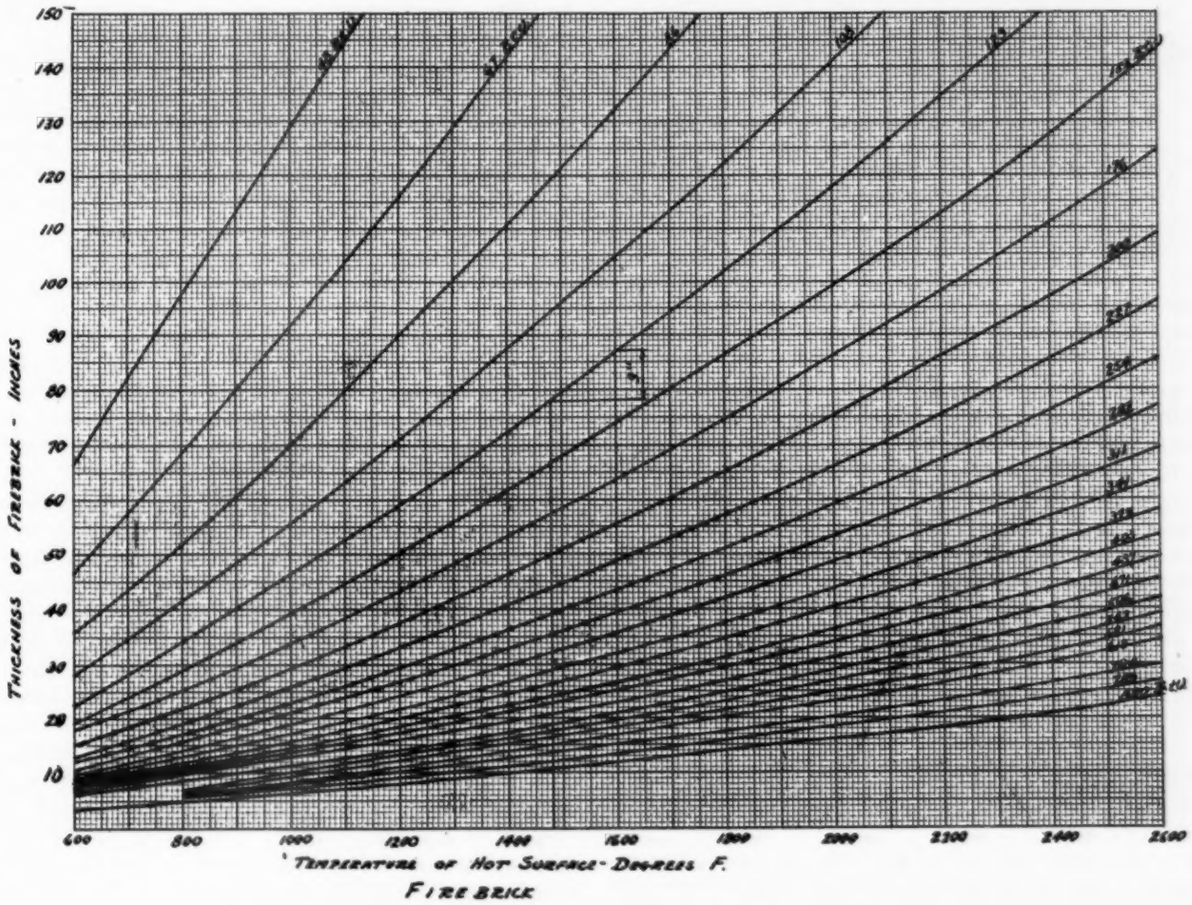



**Fig. 7—Equipotential heat curves for common firebrick**

70 F, the insulating material being ordinary firebrick, it is desired to find the thickness of the wall.

The slope of the conductivity curve, $m = \frac{8 - 5.5}{1000} = 0.0025$. From the Table, $H = 979$ Btu. Placing these values in equation (5),

$$L = \frac{0.0025}{2 \times 979}\left(1950 + 380 + \frac{2 \times 5.5}{0.0025}\right)(1950 - 380) = 13.49 \text{ in.}$$

Commercial firebrick $13\frac{1}{2}$ in. thick would be used.

EXAMPLE 3. Given a combination wall made up of two insulating materials, consisting of firebrick 9 in. thick on the hot side, and 85 per cent magnesia 2 in. thick. Referring to Fig. 6, $L_1 = 9$ in., $L_2 = 2$ in., $T_3 =$

To obtain $T_1$, use equation (3) again and $H = 86$ Btu since an equal amount of heat flows across an equal distance of the firebrick as across the magnesia in the same time. The values for $b$ and $m$ can be obtained from Fig. 5.

$$\left(T_1 + \frac{b_1}{m_1}\right)^2 = \frac{2H}{m_1}\left[L_1 + \frac{(b_1 + m_1 T_2)^2}{2Hm_1}\right] \qquad (3)$$

$$T_1 = \left\{\frac{2 \times 86}{0.0025}\left[9 + \frac{(5.5 + 0.0025 \times 476)^2}{2 \times 86 \times 0.0025}\right]\right\}^{1/2} - \frac{5.5}{0.0025}$$

$= 2788 - 2200 = 588$ F.

The curves of Fig. 7 are the equipotential heat curves

for common firebrick. The Btu value of each curve represents the emissivity loss from the cold surface. Temperatures of the hot surface are plotted against the thickness of the firebrick for the given emissivity values. Fig. 7 is used together with the Table.

HEAT LOSS FROM ONE SQUARE FOOT OF VERTICAL WALL—BTU PER HOUR, WITH AMBIENT AIR TEMPERATURE OF 70, 80, 90, 100 F

| $T_1$, F | $H$ $T_a$ = 70 F | $H$ $T_a$ = 80 F | $H$ $T_a$ = 90 F | $H$ $T_a$ = 100 F |
|---|---|---|---|---|
| 100 | 48 | 31 | 15 | 0 |
| 110 | 67 (1.9) | 50 (1.9) | 33 (1.8) | 16 |
| 120 | 86 (1.9) | 69 (1.9) | 51 (1.8) | 34 (1.8) |
| 130 | 108 (2.2) | 90 (2.1) | 71 (2.0) | 53 (1.9) |
| 140 | 129 (2.2) | 111 (2.1) | 92 (2.1) | 74 (2.1) |
| 150 | 153 (2.3) | 134 (2.3) | 115 (2.3) | 96 (2.2) |
| 160 | 176 (2.3) | 157 (2.3) | 138 (2.3) | 118 (2.2) |
| 170 | 200 (2.4) | 181 (2.4) | 161 (2.7) | 143 (2.4) |
| 180 | 227 (2.7) | 207 (2.6) | 188 (2.7) | 167 (2.4) |
| 190 | 254 (2.7) | 235 (2.7) | 215 (2.7) | 194 (2.7) |
| 200 | 282 (2.8) | 262 (2.8) | 241 (2.7) | 221 (2.7) |
| 210 | 311 (2.9) | 291 (2.9) | 270 (2.8) | 249 (2.8) |
| 220 | 341 (3.0) | 320 (2.9) | 299 (2.9) | 278 (2.9) |
| 230 | 372 (3.1) | 351 (3.1) | 330 (3.1) | 310 (3.2) |
| 240 | 403 (3.1) | 383 (3.2) | 362 (3.2) | 340 (3.2) |
| 250 | 437 (3.4) | 416 (3.3) | 395 (3.3) | 373 (3.3) |
| 260 | 471 (3.4) | 449 (3.3) | 428 (3.3) | 406 (3.3) |
| 270 | 506 (3.5) | 485 (3.6) | 463 (3.5) | 441 (3.5) |
| 280 | 542 (3.6) | 521 (3.6) | 499 (3.6) | 477 (3.6) |
| 290 | 581 (3.8) | 559 (3.8) | 537 (3.8) | 515 (3.8) |
| 300 | 619 (3.9) | 597 (3.8) | 575 (3.8) | 553 (3.8) |
| 320 | 704 (4.25) | 682 (4.25) | 659 (4.2) | 637 (4.2) |
| 340 | 789 (4.25) | 767 (4.25) | 744 (4.25) | 722 (4.25) |
| 360 | 880 (4.55) | 858 (4.55) | 835 (4.55) | 812 (4.5) |
| 380 | 979 (4.55) | 956 (4.9) | 933 (4.9) | 910 (4.9) |
| 400 | 1079 (5.0) | 1056 (5.0) | 1032 (4.95) | 1008 (4.9) |

As an illustration, consider a firebrick wall with the hot surface at 1600 F. An emissivity of 129 Btu will give a cold surface temperature of 140 F from the Table and require a wall 87 in. thick. But if a combination wall similar to that in Fig. 6 is constructed with the temperature $T_1$ still at 1600 F, $H$ again equal to 129 Btu and $L_1$, the thickness of the firebrick, equal to 9 in., $T_2$ will be given as 1480 F, as shown in Fig. 7.

Since not all insulated walls are vertical, investigations have been conducted on the heat loss from horizontal surfaces. These show that only the convection term of equation (*4*) changes. In tests, conducted in England for plates 3 to 4 ft square, the convection loss was found to be $1.29 \times 0.25\ (T - T_a)^{1/4}$ when the surface faced upward. The convection loss was $0.66 \times 0.25\ (T - T_a)^{1/4}$ when the surface faced downward. There are no reliable data for inclined surfaces. The conditions which affect the heat loss from walls are so varied and complicated that all commercial insulations are computed from values for vertical walls.

The preceding discussion has been based on the assumption that the ambient air is still. But boiler rooms are usually drafty and in recent years, some boilers, where the climate permits, have been erected outdoors. Even when the air velocity is small, data which are based on still ambient air may err considerably. Langmuir[2] has investigated the surface losses due to the effect of air velocity and has adapted the convection term of equation (*4*) to account for the heat transfer from the surface to moving air.

If $H_c = 0.25\ (T - T_a)^{1/4}$ is the emissivity due to convection in still ambient air, then for circulating air

$$H_{cv} = H_c \left(\frac{V + 68.9}{68.9}\right)^{1/3} \qquad (6)$$

where $H_{cv}$ = the rate of heat transfer from the cold surface in Btu per sq ft per hour by convection for air flowing at velocity $V$,

and $V$ = the velocity of ambient air in feet per minute over the surface.

The complete emissivity equation may then be written as

$$H = 0.155\left[\left(\frac{T + 460}{100}\right)^4 - \left(\frac{T_a + 460}{100}\right)^4\right] + 0.25\left(\frac{V + 68.9}{68.9}\right)^{1/3} (T - T_a)^{1/4} \qquad (7)$$

This agrees with equation (*4*) when $V = 0$.

When the wall under discussion is exposed to nearby hot or cold surfaces, then the calculations may be unreliable. Two boilers which are near each other, or a building wall close to a boiler wall are only two of the innumerable cases of this kind.

It is not readily understood that a bright polished surface which has a higher temperature than a dull surface may be losing heat at a much smaller rate than the dull surface. This fact is illustrated in comparing two coffee pots, one of metal and the other of earthenware. While the metal one is hotter to the touch, the coffee in the earthenware one cools sooner than the other.

[2] *Transactions* American Chemical Society, vol. 23 (1913).

## A.S.T.M. Annual Meeting

During the week of June 27 to July 1, inclusive, the Forty-first Annual Meeting of the American Society for Testing Materials will be held at the Chalfonte-Haddon Hall, Atlantic City, N. J.

An outstanding feature of the program is to be a symposium on impact testing. This is being developed jointly by the A.S.T.M. and the American Welding Society through its Welding Research Committee (jointly sponsored by the A.W.S. and the A.I.E.E.). Two sessions of the meeting will be devoted to the several papers covering specific topics. It has been developed that the impact problem consists of three related effects, (1) shape or notch effects, (2) velocity effects, (3) temperature effects. The symposium will contain papers treating each effect from the standpoint of homogeneous materials and also for welds and a discussion of the basic theory underlying all three. In addition, papers will be presented on the practical application and utility of impact testing in the prevention of failures in use.

Another session will be devoted to a series of papers on water for industrial uses. Topics to be covered include embrittlement testing of boiler waters, referee and field methods for determining dissolved oxygen in water, determination of sodium in water supplies by an indirect method, removal of dissolved salts from water by exchange filters, etc.

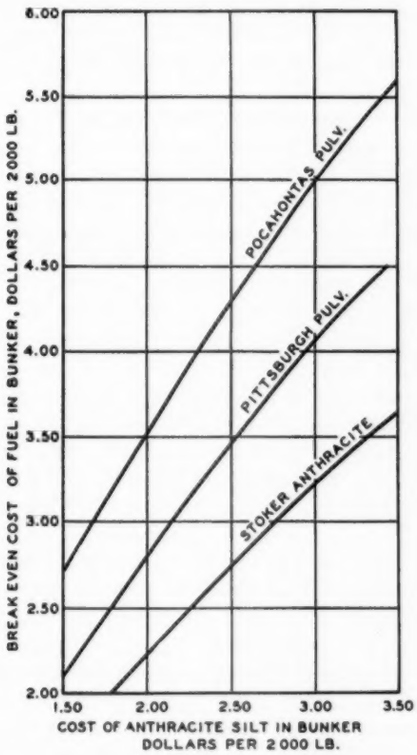
The newly organized A.S.T.M. Committee on Radiographic Testing is developing a series of papers covering pertinent topics in this field. In addition to the subjects "quantitative gamma ray radiography, radiographic testing, its uses, abuses, and limitations," and "secondary radiation in the radiography of aluminum, steel and lead," certain other topics are being considered.

Several interesting papers covering the properties of metals as affected by temperature and fatigue are scheduled.

# Pulverization of Anthracite

At the First Annual Anthracite Conference held at Lehigh University, April 28 and 29, a paper by Martin Frisch[1] discussed the pulverization of anthracite for power production, with particular reference to its economic comparison with pulverized bituminous coal and sized anthracite for stokers. The prices commanded by Nos. 4 and 5 buckwheat are too high to warrant their consideration for use in pulverized form, but the availability and prices of anthracite fines such as river coal, culm and silt, usually considered mine wastes, renders them competitive in pulverized form in certain restricted areas with intrinsically better fuels.

In making such comparisons, not only must the fuel cost be considered but also fixed charges on the firing




**Comparative fuel cost for equal economy with a 100,000-lb per hr unit**

equipment and costs of power and maintenance; for there is a fixed cost as well as an operating cost handicap against the anthracite mills which must be offset by a lower purchase price per ton of fuel fired. The chart shows the approximate "break even" prices for a 100,000-lb per hr unit between anthracite silt, pulverized Pocahontas coal, high-volatile Pittsburgh coal and a mixture of 75 per cent No. 3 and 25 per cent No. 4 buckwheat burned on a stoker.

The high ash and low volatile content, and the consequent low velocity of flame propagation makes fine pulverization and easy control of the primary air-coal ratio leaving the burner essential with pulverized anthracite in order that ignition and self-sustaining combustion may be maintained under rapid and wide load swings. To obtain the required steam output at the necessary fineness calls for mills considerably larger than those required for the same steam output with bituminous coal, because the grindability and heating value of the anthracite, as received, are lower and the moisture and ash content higher. The table gives such comparisons for a maximum continuous output of 100,000 lb of steam per hour.

In order that pulverized high-ash, low-volatile anthracite may be burned without the addition of other richer fuels to support combustion, it is necessary to remove and vent to the furnace some of the air from the mixture leaving the mill so that the mixture delivered to the furnace by the burner will contain much less air than is usual with pulverized bituminous coal. The maximum permissible amount of air per pound of anthracite fired may be only one-tenth to one-sixth of that for bituminous coal. This follows from the fact that the velocity of flame propagation through air suspension of pulverized anthracite is from one-tenth to one-eighth of that attained in suspension of bituminous coals.

Mr. Frisch mentioned two paper mill installations now successfully burning anthracite silt, one of which

COMPARISON OF PULVERIZED ANTHRACITE SILT WITH BITUMINOUS COALS

| Pulverized coal fired | Anthracite | Bituminous | |
|---|---|---|---|
| Grade | Silt | High Vol. | Low Vol. |
| Seam or District | Schuylkill | Pittsburgh | Pocahontas |
| Proximate Analysis: | | | |
| Volatile | 7.0 | 32.3 | 17.6 |
| Fixed Carbon | 70.4 | 55.6 | 73.0 |
| Moisture | 10.6 | 5.5 | 4.2 |
| Ash | 12.0 | 6.6 | 5.2 |
| Btu/lb | 11,500 | 13,500 | 14,000 |
| Grindability: | | | |
| ASTM D-408-35 Rev. for 80% through 200 mesh | 1350 | 1077 | 627 |
| ASTM D-409-35, per cent | 50 | 66.5 | 100.5 |
| Relative capacity at same fineness | 0.56 | 0.70 | 1.00 |
| Required fineness, per cent through 200 mesh | 85 | 70 | 75 |
| Boiler efficiency—same boiler, no heat recovery except for preheating primary air | 77 | 80 | 80 |
| Fuel required, lb/hr | 12,200 | 10,000 | 9650 |
| Size of pulverizer as measured by its capacity on anthracite at the fineness required for anthracite, lb/hr | 12,200 | 5600 | 4100 |
| Approx. cost of pulverizing and firing equipment | $34,000 | $24,000 | $22,000 |
| Power consumption, kwhr/ton | 35 | 15 | 12 |
| Maintenance cost, ¢/ton | 4 | 2 | 1 |

was placed in service in 1935 and the other in 1937. The anthracite fines in these cases, which are pulverized in ball mills, vary between $2\frac{1}{2}$ and 8 per cent volatile, 10 and 17 per cent moisture, and 10 and 20 per cent ash content; no predrying is employed. Wide load swings ranging between 19,000 and 75,000 lb of steam per hour are encountered. The power consumption of the mills at normal capacity averages about 35 kwhr per ton and the total pulverizing cost is about 20 cents per ton including maintenance.

[1] Chief Engineer, Boiler and Pulverizer Division, Foster Wheeler Corporation, New York.

# When an Engineer Is Liable for Defective Plans and Specifications

By LEO T. PARKER
Attorney at Law
Cincinnati, Ohio

**By quoting pertinent court decisions dealing with various phases of this question and commenting thereon, the author shows that the engineer is not required to submit perfect plans and specifications but that they must be specific and within the standards prescribed by law. He may not be responsible for changes ordered by the owner, but unfamiliarity with appliances and equipment specified cannot be accepted as an excuse. Moreover, where the engineer is engaged to inspect and to supervise construction he cannot escape responsibility for defects that develop.**

OCCASIONALLY engineers are confronted with the embarrassing predicament of having unintentionally furnished plans and specifications which prove defective in a greater or lesser degree. Of course, if the mistake is of minor importance the defect often may be remedied without difficulty. However, occurrences are frequent where plant owners have endeavored to hold engineers responsible for furnishing plans and specifications that are only slightly defective, particularly if unfriendly relation arises between the parties.

On the other hand, where the defects are of major importance, the plant owner may feel justified in seeking legal relief in holding the engineer responsible for the unnecessary expenditures thus incurred.

It is the purpose of this article to review the leading higher court cases which involve various phases of the law relating to plans and specifications which have minor defects, as well as those that are unusually defective.

### *Legal Duty of Engineer*

It is important to know that an engineer is not legally required to prepare *perfect* plans and specifications, provided he does not guarantee to the plant owner that his work will be above the standard required by law. In other words, the law is well established that an engineer is liable for supplying imperfect plans and specifications *only* where it is proved to the satisfaction of the court that he did not exercise that degree of care which ordinarily is used by other experienced and reasonably proficient engineers.

For instance, in the leading case of Chapel v. Clark (*76 N. W. 62*), it was disclosed that a plant owner discharged an engineer and refused to pay the agreed compensation on the grounds that the latter had prepared faulty and defective plans and specifications. After thoroughly considering all details of the case the court found that the plans and specifications were reasonably perfect and held the plant owner liable for damages and payment of the full amount of the agreed compensation. This court explained the law on this subject, in the following language:

"A person who holds himself out to the public in a professional capacity holds himself to be possessed of average ability in such profession, and the law implies that he contracts with his employer (1) that he possesses that requisite degree of learning and experience which is ordinarily possessed by the profession in the same art or science and which is ordinarily regarded by the community, and by those conversant with that employment, as necessary and sufficient to qualify him to engage in such business; (2) that he will use reasonable and ordinary care and diligence in the exercise of his skill, in the application of his knowledge, to accomplish the purpose for which he is employed; (3) in stipulating to exert his skill and apply his diligence and care, he, like other professional men, contracts to use his best judgment . . . . . When he possesses the requisite skill and knowledge, and in the exercise thereof has used his best judgment, he has done all that the law requires."

On the other hand, it is the recognized legal duty of an engineer to obtain from the plant owner all facts necessary to enable him to prepare proper and usable plans and specifications for a proposed project. If he fails to obtain this information, he cannot later excuse himself for failure to do so when controversy arises over the plans and specifications being incompletely prepared or defective.

For example, in Nave v. McGrane (*113 Pac. 82*), it was shown that an engineer prepared plans and specifications which contained no form of contract, or bid, or bond for the construction of the building. The owner rejected the work because the plans did not contain complete information in regard to the details and they did not contain the date of beginning work, the date of completion, the time or manner of payment, whether work should be continuous or intermittent, nor the amount and character of indemnity against personal injury or against liens.

In holding this engineer *not* entitled to compensation for the plans and specifications, it is interesting to observe that the court said:

"The plans, specifications, contracts, and bond should

be so specific that there could be no valid controversy as to the kinds and quality of materials to be used, or the workmanship, the time when the construction should begin and when completed, and the other matters above stated."

*Excuses Not Acceptable*

An engineer is required by the law to be familiar with all of the appliances and equipment that he includes in the plans and specifications. Therefore, he cannot avoid legal liability for imperfect plans and specifications on the plea that he was not an expert in equipment that he incorrectly specifies.

For illustration, in the leading case of Hubert v. Ailken (*15 Daly, 237*), an engineer attempted to avoid liability for improperly specifying special equipment on the contention that he was not an expert on these particular appliances. The court held him liable in damages for the resultant defective structure, saying:

"He is an expert in cements, in mortar, in the strength of materials, in the art of constructing the walls, and he is in duty bound to possess reasonable skill and knowledge as to all these things."

*Plans Changed by Owner*

If an engineer had prepared reasonably good plans and specifications and the plant owner authorized various changes as the work progressed, the law does not require the modified portions of the plans and specifications to be as perfect as the original.

This point of the law was thoroughly considered by the higher court in Kortz v. Kimberlin (*165 S. W. 654*). In this case a plant owner attempted to hold an engineer liable in damages amounting to several hundred dollars for defects in numerous items which the owner had ordered changed from the plans and specifications as originally submitted. The court held the engineer not liable for damages resulting from the defects and explained that numerous changes ordered from the original plans by the owner are likely to confuse the engineer. Also, in this same case, the owner attempted to strengthen his suit on the contention that the engineer had not exercised care in properly supervising the work to know that defective materials were being used.

The court held the engineer not liable on this charge and explained the law as follows:

"While it is true that a public profession of an art is a representation or undertaking to all the world that the professor possesses the requisite ability and skill, and it therefore follows that he is presumed to possess the skill and ability necessary to the practice of his profession and is liable for damages occasioned by defective plans, yet he does not undertake that his plans will be absolutely perfect, and is liable only for a failure to exercise reasonable skill in the preparation of the plans. Furthermore, if he be employed to superintend he is not liable at all hazards for every defect but is only required to use reasonable care and diligence in seeing that the work is properly done. The mere fact, therefore, that some of the material is defective, or that some of the construction work is not done in a workmanlike manner is not sufficient to establish as a matter of law that he has not fully performed his contract. Under such circumstances, the question whether or not he used reasonable care and diligence in superintending the work is a question for the jury to decide."

*Sufficiency of Plans*

It has been held in numerous cases that an engineer impliedly agrees to supply reasonably perfect plans and specifications and upon his absolute failure to do so, the plant owner may legally refuse to compensate him for any services rendered. If, however, only minor details of the plans are imperfect, a different situation arises.

Therefore, frequently it is advantageous for an engineer and owner to be informed of an equitable method of determining a fair amount deductable from the engineer's agreed compensation, where the plans and specifications are defective, without entering into court procedure. Obviously, it is beneficial for the parties to amicably settle a dispute of this nature, instead of entering into expensive litigation.

In Johnson v. Wanamaker (*17 Pa. Super. 301*) it was held that if the plans simply are incomplete, requiring only these rvices of an ordinary draughtsman to perfect the details, then the owner may legally deduct from the agreed compensation only the cost of making the corrections.

On the other hand, it has been held that if the plans contain material defects which only the skill of an experienced engineer can discover, the engineer must pay for correcting the plans to eliminate defects; and, if the plans are so defective as to be incapable of meeting the requirements for which they are intended, they may be utterly worthless.

*Plans Must Be Lawful*

Another common source of litigation is where an engineer prepares plans and specifications not strictly in accordance with certain existing laws. For instance, in Straus v. Buchman (*89 N. Y. S. 226*), the plant owner sued for damages because certain plans and specifications were in violation of the express provisions of the building laws.

The court held the owner entitled to a recovery, saying:

"It was the duty of the defendants (engineers) under their contract not only to see that the beams were properly placed, but especially to see that the placing of them conformed to the requirements of the statute. This they failed to do."

Also, it is important to know that an engineer may not avoid liability for failure to prepare plans and specifications in accordance with the laws, by simply referring to certain laws in the specifications. In other words, an engineer is required to specify and show all necessary lawful details with respect to the materials and appliances to be used.

The law is well established that an engineer always is required to exercise ordinary care and good judgment, as well as lawful requirements, when preparing plans and specifications.

For illustration, in Trunk & Gordon v. Clark (*145 N. W. 277*) it was shown that an engineer entered into an agreement with an owner to furnish the necessary plans and specifications and to make inspections of the work during its progress. Litigation developed when the owner refused to pay the agreed amount of the bill and

contended that the plans and specifications were so negligently prepared as to be entirely worthless, and furthermore, that the engineer had agreed to inspect the work to see that it was being done lawfully but that he failed to do so.

The court held the engineer not entitled to compensation because he had not prepared the plans and specications with a reasonable degree of technical skill and had also neglected to make proper inspections during the progress of the work.

### *Responsibility for Superintendence*

Moreover, an engineer who superintends the construction of work, generally, is responsible for unusual defects which develop. For instance, in Schreiner v. Miller, (*24 N. W. 738*), it was disclosed that an engineer entered into a contract with an owner to prepare plans and specifications and superintend the construction of work. After the work was completed, the owner refused to compensate the engineer, contending that the latter had furnished defective plans and had permitted a workman to install defective materials. The court held that since the engineer had superintended the construction of the building and had done so negligently, he was responsible for defective work.

On the other hand, a plant owner is not permitted to deprive an engineer of full compensation on the contention that the completed work is slightly defective. In other words, where an engineer's completed work is only slightly defective, he is entitled to recover full compensation less the cost of repairing the defects.

# EQUIPMENT SALES

## Boiler, Stoker, Pulverized Fuel

**as reported by equipment manufacturers of the Department of Commerce, Bureau of the Census**

### Boiler Sales

| | 1938 Water Tube | | 1937 Water Tube | | 1938 Fire Tube | | 1937 Fire Tube | |
|---|---|---|---|---|---|---|---|---|
| | No. | Sq Ft | No. | Sq Ft | No. | Sq Ft | No. | Sq Ft |
| Jan. | 52 | 201,151 | 52 | 256,368 | 35 | 42,752 | 65 | 84,889 |
| Feb. | 48 | 185,257 | 51 | 198,957 | 45 | 55,173 | 74 | 89,133 |
| Mar. | 61 | 254,427 | 142 | 791,168 | 50 | 49,039 | 150 | 211,733 |
| Jan. to Mar. Inclusive | 161 | 640,835 | 245 | 1,246,493 | 130 | 146,964 | 289 | 385,755 |
| 1937—12 mos. | | | 861 | 4,058,481 | | | 941 | 1,178,805 |

### Mechanical Stoker* Sales

| | 1938 Water Tube | | 1937 Water Tube | | 1938 Fire Tube | | 1937 Fire Tube | |
|---|---|---|---|---|---|---|---|---|
| | No. | Hp | No. | Hp | No. | Hp | No. | Hp |
| Jan. | 28 | 9,484 | 63 | 25,278 | 76 | 10,991 | 140 | 21,636 |
| Feb. | 36‡ | 12,450‡ | 45 | 16,591 | 76‡ | 12,216‡ | 120 | 20,650 |
| Mar. | 54 | 18,820 | 80 | 38,074 | 52 | 9,434 | 179 | 24,709 |
| Jan. to Mar. Inclusive | 118 | 40,754 | 188 | 79,943 | 204 | 32,641 | 439 | 66,995 |
| 1937—12 mos. | | | 659 | 262,834 | | | 2,628 | 361,346 |

* Capacity over 300 lb of coal per hour
‡ Corrected

### Pulverizer Sales

| | 1938 Water Tube | | | 1937 Water Tube | | | 1938 Fire Tube | | | 1937 Fire Tube | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | No. | | Cap. Lb | No. | | Cap. Lb | No. | | Cap. Lb | No. | | Cap. Lb |
| | N.† | E.† | Coal/Hr | N.† | E.† | Coal/Hr | N.† | E.† | Coal/Hr | N.† | E.† | Coal/Hr |
| Jan. | 5 | — | 40,500 | 35 | 7 | 554,900 | 1 | — | 1,000 | — | 2 | 1,700 |
| Feb. | 7 | 1 | 38,020 | 2 | 6 | 68,300 | — | 1 | 800 | — | 4 | 3,600 |
| Mar. | — | 2 | 26,100 | 59 | 3 | 713,440** | — | 1 | 700 | 1 | 1 | 2,000 |
| Jan. to Mar. Inclusive | 12 | 3 | 104,620 | 96 | 16 | 1,510,640** | 1 | 2 | 2,500 | 1 | 7 | 7,300 |
| 1937—12 mos. | | | | 214 | 65 | 2,924,590 | | | | 3 | 10 | 11,900 |

† N—New boilers E—Existing boilers.
** Revised.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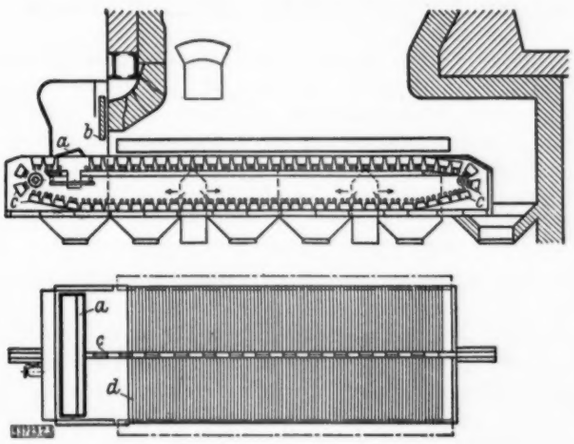
# STEAM ENGINEERING ABROAD

## As reported in the foreign technical press

### Steinmüller Pusher Grate

The well-known firm of Steinmüller in Germany has brought out a new stoker, or pusher grate, recently described in *Zeitschrift des Vereines deutscher Ingenieure* (Vol. 81, No. 47). As shown in the accompanying sketch the design is simple. A pusher of triangular section, which extends the width of the grate and normally rests beneath the green coal, is moved from front to rear of the grate and back by a centrally-located chain at intervals of 1 to 3 min. Because of its shape the rearward motion of the pusher carries more fuel toward the back of the grate than does the forward motion, although




**Plan and elevation of Steinmüller pusher grate**
*a* represents the pusher, *b* the grate, *c* the movable grate chain and *d* the grate bars which are stationary

both produce agitation of the fuel bed. By varying the shape and travel of the pusher the requirements of different coals may be met.

Zoned air under forced draft may be provided if desired. In that case when burning coals that tend to form floating coke, the underfire air admission is throttled to the zones to reduce the fines in floatation and the secondary air over the fire is proportionately increased to mix with and burn the excess gases given off while the fuel is being agitated.

Tests have demonstrated that the stoker is capable of burning a wide variety of coals.

### Grid Operations for 1937

The report of the Central Electricity Board (Great Britain) for the year 1937 has lately been released and is reviewed in the April 1 issue of *Engineering*. During 1937 the total output amounted to nearly 23 billion kilowatt-hours, an increase of approximately 12 per cent over 1936. At present the Grid System comprises 4180 miles of which 2938 miles are operated at 132 kv. The number of selected stations was 137 with a total installed capacity of 7,653,370 kw; in addition to which there were 25 stations working under temporary agreements. Of these 162 stations, only 21 ran for the full year, 27 ran between 6000 and 8760 hr, 65 between 2400 and 6600 hr, and 42 for less than 2400 hr; the remaining 7 were shut down. It is estimated that the total saving arising from reduction in reserve capacity through the Grid System of interconnection to date has amounted to the equivalent of about 85 million dollars.

During 1937, 557,000 kw of new generating capacity and boilers with an aggregate capacity of 5,349,500 lb of steam per hour were brought into service, the installation of an additional 217,000 kw having been delayed through the congestion caused by the re-armament program.

### Operating Experiences with Large Benson Boilers

*The Steam Engineer* (London) for April 1938 reviews a recent paper by Dr.-Ing. H. Lent before the Vereines deutscher Ingenieure relating experiences with three comparatively large Benson boilers at the Scholven power plant of Bergwerksgesellschaft Hibernia A.-G. This chemical works has a 412-lb plant topped by a high-pressure installation consisting of three Benson boilers, operating at 1777 lb pressure, 986 F steam temperature and each capable of a sustained output of 200,000 lb of steam per hour. These units, which are fired by Krämer mills, have been in service about a year and a half and furnish steam to high-pressure turbines. Two additional boilers of like capacity are now being installed.

The availability factors reported for the three boilers were 75, 77 and 92 per cent, respectively. The lower availability of the first unit installed was due primarily to outages caused by tube failures in the radiation and superheater sections and that of the second unit to the fact that experiments and tests were confined largely to it. The tubes of the radiation section are 1 in. and $1^1/_4$ in. and all connections are gas welded without annealing. The superheater tubes are of Sicromal-CS 65 steel and the failures appeared to be traceable to gas inclusions in the ingots, lack of uniformity in annealing and to the notch effect of decarbonized coarse grain zones. Numerous leakages appeared in the economizer welds.

The heat absorption in the radiation section of the first boilers was about 10 per cent greater than assumed in the design, hence the gas temperature behind the air heater was low (only 248 to 284 F) instead of the expected 464 F. To rectify this, part of the radiant heating surface was covered with refractory material, and half of the economizer surface was removed. Finally,

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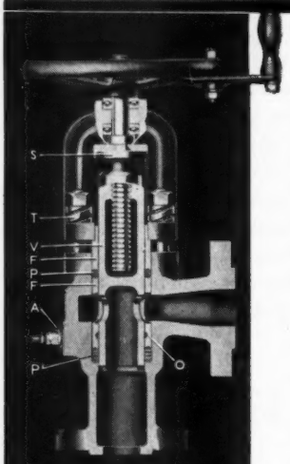


Yarway Seatless Blow-off Valve. Operation: After Valve is closed, shoulder S on plunger V contacts with upper follower gland F, forcing it down into body and compressing packing P above and below port. Annular groove O connects with Alemite fitting A for lubricating plunger and packing. Yoke springs T maintain continuous pressure through follower gland F on packing rings P.

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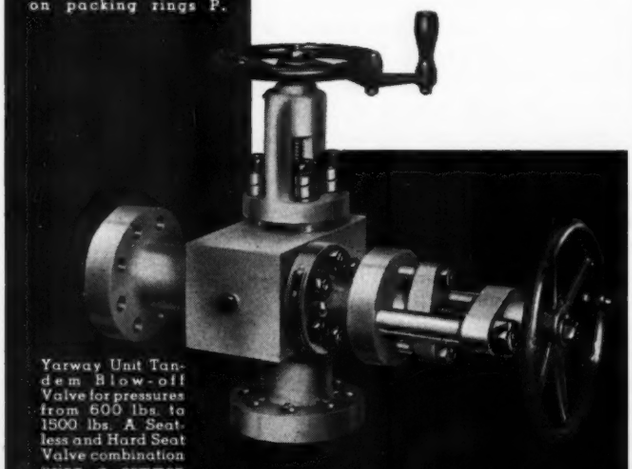

Yarway Unit Tandem Blow-off Valve for pressures from 600 lbs. to 1500 lbs. A Seatless and Hard Seat Valve combination using a common forged steel body.

YARWAY

BLOW-OFF VALVES


by removal of the entire economizer the desired flue gas temperature was then attained. No refractory appears available that will stand up well to the slag under the conditions of service imposed.

The Krämer mill firing is reported as satisfactory with the 22 per cent ash and 17 per cent moisture coal that is burned. The only difficulty with the mills was excessive hammer wear, despite the employment of manganese steel for the hammers. These lasted from 700 to 1000 hr. Later pearlite cast iron was substituted and increased their life to 1500 to 2000 hr. On test, with the economizer removed, one unit showed an efficiency of 81 per cent with 9050 Btu coal and a 14 per cent flue gas loss. The corresponding furnace heat release was 18,500 Btu per cu ft.

No automatic combuston control is employed and regulation is based on the superheat temperature. Starting from a cold condition, the units can be brought up to 75 per cent of full load in 20 min. Salt deposits in the turbines have been less than anticipated.

## Temperature Measurements of Gases

In *Archiv für Wärmewirtschaft und Dampfkesselwesen* of January 1938, Dr. Ing. H. Jaeger shows that temperature measurements of gases differ basically from those of fluids, vapors and superheated vapors, and that accurate measurement in the case of gases is impossible without removing the influence of radiation.

In the case of fluids and vapors, also superheated vapors, the difference between the temperature of the substance being measured and that of the measuring device becomes smaller the larger the heat transfer coefficient, which in such cases is usually large. Furthermore, the surface and mass of the measuring device is usually kept as small as possible. Hence, the errors are generally small and are minimized by providing sufficient insulation.

However, for gases (except $CO_2$) which permit the passage of radiation, the coefficient of heat transfer is usually small and the radiation often large. Thus a temperature is measured which falls between that of the gas and that of the wall. This approaches the wall temperature the smaller the difference. Moreover, surfaces, when subjected to both high temperature and gases that attack the metal, will become so affected that the radiation factor may rise to tenfold or more and the usual form of shielding becomes ineffective.

Radiation losses may be cancelled if the surrounding walls are well insulated and electrically heated to the gas temperature, or where this is impossible, the measuring device may be surrounded by an electrically heated shield. The walls, however, must not be hotter than the gas.

Where the gas to be measured contains $CO_2$ and water vapor, the gas radiation portion must be subtracted from the heat transfer coefficients in order to obtain the true convection coefficient.

The influence of velocity and gas temperature upon the heat transfer when using a suction pyrometer are such that the transfer coefficient increases as the fourth power of the temperature difference and also directly as the gas velocity.

## Electrostatic Dust Precipitators in British Stations

Eight or nine years ago, says *The Fuel Economist* (London), there was only one electrostatic dust precipitator installed in a central power station in England and practically none in stations controlled by British interests outside of England. Conditions have now changed considerably as is shown by a tabulation of twenty such installations in well-known central stations.

Various arrangements, as employed in these stations, are shown. In some of these the fans are placed following the precipitator in order to have the fans handle clean gas and thus cut down maintenance. Despite the increased suction under which the precipitators operate in these circumstances, no trouble has been experienced. In this connection mention is made of the three precipitators at the Kirkstall Station at Leeds where the suction is in the neighborhood of 7 in. of water. A tabulation shows the current consumptions of these precipitators, which are mostly of two well-known makes, to be proportional to the size and the efficiency of dust removal. That is, the larger the gas volume, the more electrodes must be used, and the higher the efficiency the greater the power consumption. The operating voltage ranges from 40 to 42 kv and the current from 110 to 208 milliamp.

## Suggested Bombproof Power Station

While no actual bombproof power installations have been made in England, such as has been done on the Continent, nevertheless, interest in defense against aerial attack has reached the point where such plans are being offered. In a recent paper before the Institution of Structural Engineers, A. J. Hodgkinson presents such a design which is reproduced in the April 15 issue of *The Engineer* (London). The suggested plan is for a station of 50,000 kw, the building being somewhat dome-shaped to deflect a bomb and having a double shell, each wall of which is of 5-in. reinforced concrete separated by a considerable space. Barrier walls are also provided to isolate damage and there is a bombproof shelter for the operating crew. It is suggested that while such a building might be more vulnerable to attack than an underground installation, the latter would be much more costly.

## New Danish Power Station

A new steam power station of 68,000 kw initial capacity, with provision for 50,000 kw additional capacity, has lately been erected to cover peak-load requirements for Seeland, the principal island of Denmark. This station will operate for about 3000 hr per year and the average condensing water temperature is 50 F, thus insuring operation at a high vacuum. For this reason the last row of blades in the low-pressure stages of the turbines is specially air-hardened to resist erosion from drops of water and drainage channels are provided. The two 34,000-kw, 3000-rpm single-cylinder turbine-generators, built for a throttle pressure of 500 lb and 800 F, are described in the March issue of *The Brown Boveri Review*.

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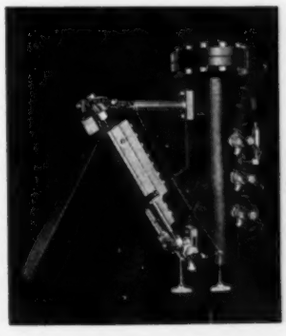

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# REVIEW OF NEW BOOKS

**Any of the books reviewed on this page may be secured from Combustion Publishing Company, Inc., 200 Madison Ave., New York**

## Elements of Thermodynamics Second Edition

**By Ernest M. Fernald**

The author, who is Professor of Mechanical Engineering at Lafayette College, has arranged the material primarily for university courses in thermodynamics, avoiding many of the more complex aspects of the subject and adapting the treatment to engineering applications. Examples are employed freely to demonstrate practical applications. Most of the text of the first edition has been rewritten and new chapters have been added on mixtures, semi-perfect gases, combustion and fundamental relations which are developed symmetrically.

In certain respects the treatment differs from that in most books on thermodynamics, notably with reference to entropy and to moist air, dry air and water, which are handled separately.

The book contains 330 pages, 6 × 9 in. Price $3.50.

## Air-Conditioning Furnaces and Unit Heaters

**By J. R. Dalzell**

As may be implied from the title, much of the text deals with domestic installations although considerable space is also given to industrial applications. The treatment is elementary and practical. Among other things, it tells how to convert old heating systems into air-conditioning systems; how to air condition new houses; how to figure summer and winter air conditioning; how to select apparatus, to figure pipe and duct sizes; how to dehumidify in summer and humidify in winter; and how to figure unit heaters. Control equipment is also discussed at length. Typical plans, tables and a psychrometric chart are included.

The book contains 430 pages, $8\frac{1}{2} \times 5\frac{1}{2}$ in., fully illustrated. Price $3.

## The Practice of Lubrication

**By T. C. Thomson**

This is the third edition, the first having been brought out in 1920, since which time progress in the industry has made the refiner less dependent upon the nature of the crude oil. The author is a Danish consulting lubricating engineer with a background of many years' experience as chief engineer of the Vacuum Oil Company Ltd. (London) and as research engineer with the Anglo-Mexican Petroleum Company Ltd.

The present book discusses basically mineral oils, fixed oils and fats, semi-solid lubricants and the laws of friction. Over fifty pages are devoted to testing lubricants. The proper lubrication for various types of bearings is treated at length and separate chapters are devoted to the lubrication of steam turbines, steam engines, oil and gas engines, compressors, refrigerating machinery, transmission shafting, machine tools, textile machinery, rolling mills and railway rolling stock. The subject is treated in a most practical and useful manner. There are 838 pages, 6 × 9 in., illustrated. Price $6.

## Air Conditioning

**By Richard E. Holmes**

This book covers the subject of air conditioning in winter and summer from a practical standpoint and is based largely on a series of lessons given by the author at the Westinghouse works to students preparing for work in the air conditioning industry. Among the topics dealt with in the various chapters are, nature of the atmosphere, psychrometry, requirements for comfort, descriptions of various air conditioning systems, air conditioning loads as influenced by air infiltration and heat-transmission loss, types of heaters, humidifiers and air filters, condensing equipment and temperature and humidity controls.

Much tabular data, resulting from the research work of the American Society of Heating and Ventilating Engineers, are included as well as tables giving the properties of refrigerants.

There are 296 pages, 6 × 9 in. Price $3.

## Handbook of Arc Welding

This is the fifth edition of a 1012-page book prepared by the Lincoln Electric Company dealing with design, practice and procedure involved in arc welding. Among the twelve hundred illustrations are many examples of welded machine designs and standard shapes employed in welded structural construction. It contains a wealth of data for the use of designing engineers and architects as well as for welding operators and supervisors.

Bound in leatherette and 6 × 9 in. size, the price of the book is $1.50.

## Sheet Metal Work

**By William Neubecker**

This is a book of 360 pages and over 400 illustrations written primarily for the draftsman engaged in laying out sheet metal work. Geometrical principles and the development of complex surfaces are dealt with and considerable space is given to duct work. The treatment is strictly practical.

The price is $2.50.

# The Effect of Copper on Alloy Steels

In a paper before the Annual Meeting of the Iron and Steel Institute (Great Britain), May 4 to 6, R. Harrison gave the results of some recent investigations on the mechanical properties of a number of alloy steels containing from 1 to 6 per cent copper, in the oil-hardened and tempered condition and also in the temper-hardened condition. The maximum increase in hardness, due to temper-hardening, was obtained in a low-carbon steel containing 0.5 per cent of chromium and 1.6 per cent of copper. Increasing the alloy content in order to obtain a higher final hardness introduced a tendency to air-hardening, and the temper-hardening effect was gradually masked by the softening due to the normal tempering.

The highest tensile strength obtained by temper-hardening alone—that is, in the absence of any air-hardening during normalizing—was slightly over 50 tons per square inch. The low Izod impact figures usually associated with the temper-hardening process were considerably improved by increasing the reheating time or the temperature.

The chief advantage of temper-hardening is the simplicity of the treatment which, however, can only be secured at a sacrifice in some of the properties attainable; another arrangement for the temper-hardening is that it should minimize the mass effect, but this advantage does not apply in sections of moderate size, since copper itself has a beneficial effect in increasing the depth of quench-hardening. By temper-hardening alone, without the aid of some air-hardening in normalizing, a tensile strength of over 50 tons per square inch would be difficult to obtain.

Although temper-hardening may provide material of good properties when the composition and the treatment are carefully chosen, and may prove useful in particular cases, it was the author's opinion that this is not likely to displace the usual method of oil-hardening and tempering for the ordinary materials employed in construction.

Copper may, however, prove to be a useful alloy element in steels to be used in the hardened and tempered condition. Considerably higher tensile properties were obtained in a medium-carbon steel containing 1 per cent copper than in a similar steel containing 1.4 per cent nickel; and the properties of 3.4 per cent copper, 0.9 per cent chromium steel were slightly better than those of a 3.5 per cent nickel, 0.9 per cent chromium steel.

The investigation indicated that if difficulties in mechanical treatment results from the limitation of the forging temperature necessitated by higher copper contents, a steel containing nickel and copper, free from these limitations, may be employed without detriment to the mechanical properties.


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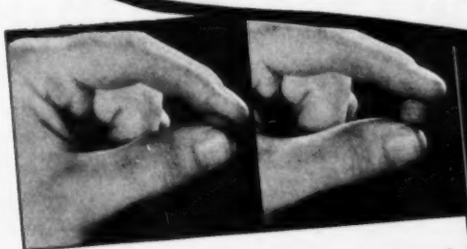

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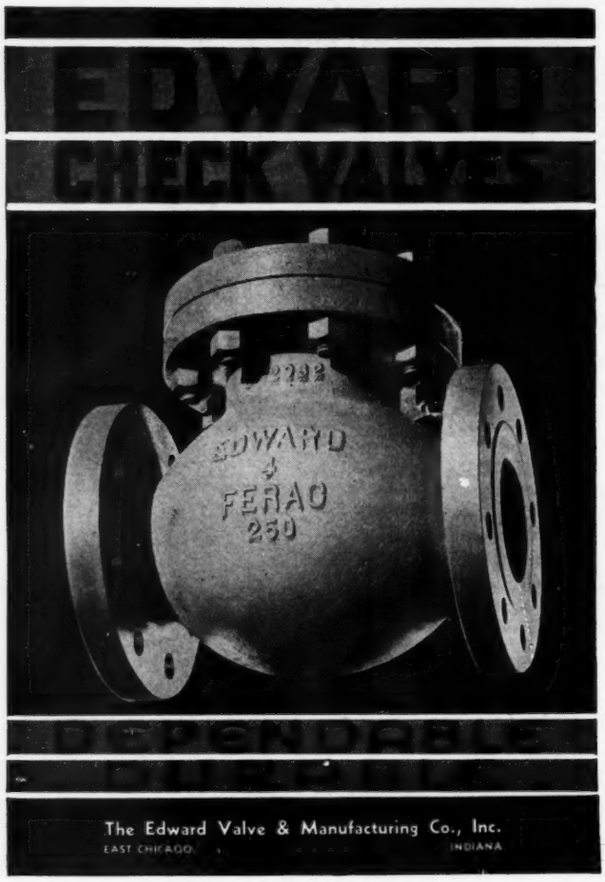


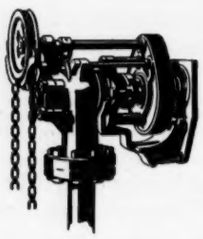

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. . . are readily meeting high temperature conditions without makeshift in design, material or parts. The sound and proven principles of the Vulcan valve operating head, bearings and elements guarantee operating economies which make both the annoyance and cost of frequent servicing unnecessary.

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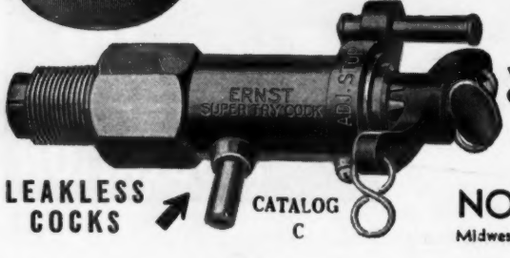


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WATER GAGES WILL FIT ANY TYPE OF WATER COLUMN OR BOILER FOR ALL PRESSURES
SIMPLE AND EASY TO INSTALL A GAGE GLASS
NO WRENCHES OR TOOLS REQUIRED
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ALIGNMENT AND STIFFENER ROD
NO CAGE GLASS PACKING NUTS
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